DOE/NASA CONTRACTOR REPORT

DOE/NASA CR-150576

# PROTOTYPE SOLAR HEATING AND COOLING SYSTEMS INCLUDING POTABLE HOT WATER (Quarterly Report)


Prepared by

Solaron Corporation
Solaron Energy Systems
Denver, CO 80222



Under Contract NAS8-32249 with

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama 35812

For the U. S. Department of Energy





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15. SUPPLEMENTARY NOTES

This work was done under the technical management of Mr. Mitchell Cash, George C. Marshall Space Flight Center, Alabama.

16. ABSTRACT

This report covers the progress made in the development, delivery and support of two prototype solar heating and cooling systems including potable hot water. The system consists of the following subsystems: collector, auxiliary heating, potable hot water, storage, control, transport, and government-furnished site data acquisition. Included is a comparison of the proposed Solaron-Heat Pump and Solaron-Desiccant Heating and Cooling Systems, Installation Drawings, data on the Akron House at Akron, Ohio, and other program activities from July 1, 1977 through November 9, 1977.

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## Part I - Summary

1.0 This report will summarize and bring up-to-date the reporting on this contract for the period from 1 July 1977 through 9 November 1977. The following information, as applicable, is included: Contract Changes, Schedules, and Technical Performance.

## Part II - Contract

2.0 Contract Changes - There have been no contract changes during the reporting period. There is, however, a Contract Change Proposal, initiated at the end of the reporting period, which is subject to NASA review and approval.

## Part III - Schedules

3.0 The latest revision to the schedule was submitted with the August monthly report. The schedule is, however, subject to review and revision depending on the final selection of the type of system to be installed in the demonstration sites, whether desiccant or heat pump. A new schedule will be prepared upon finalization of this decision. A proposed change of scheduled program milestones (Para. 2.0 of SHC 3014) will be submitted with the Change Proposal identified in Part II above.

## Part IV - Technical Performance

4.0 The technical performance under this Contract is reported under separate headings as follows:

(4.1) - Desiccant System - Solaron Engineering Design.

(4.2) - Desiccant Systems - University of Wisconsin Subcontract.

(4.3) - Heat Pump Systems - Carrier Subcontract.

(4.4) - Installation Planning - Akron House.

The content is essentially a composite and summary of the technical performance as reported in the July through October 1977 monthly reports, but in addition to this it brings the technical status of the contract up to date through the Preliminary Design Review (PDR) which was held at Solaron, Denver on 9 November 1977.

4.1 Desiccant System - Solaron Engineering Design

4.1.1 It had been determined prior to this reporting period that the Recirculation Cycle, Option 1 was the optimum cycle for the Akron Site, considering both the cycle efficiency and the capital cost investment. Subsequent to this decision, the Wisconsin computer simulation was fine tuned for the specific system. This is reported in more detail in para. 4.2.

4.1.2 A Technical/Quarterly Review Meeting was held at Solaron, Denver on 30 August 1977, with Messrs. Mitchell Cash and Bill Richardson representing NASA, and Dr. George Löf and Messrs. Les Shaw, John Meeker, Rod Oonk and Don Bloomquist representing Solaron. Solaron reviewed the entire program up-to-date, and discussion followed relative to the program schedule and plans for testing.

Solaron agreed to the following:

1. To generate a test plan.

2. To revise the schedule based on an "installation complete" date in the last half of May 1978, and PDR based on the test plan.

3. Up-dated heat pump data to be included in the monthly report for August.

4. A psychrometric presentation of desiccant system performance when operating at "off-peak" conditions.

5. A planned funding curve for the balance of the Contract.

These items were all completed and conveyed to NASA. The Test Plan is included with this report as Attachment I. The PDR was scheduled and held on 11/9/77. The data package for the PDR, sent to NASA prior to the meeting, was entitled "Comparisons of Proposed Solaron - Heat Pump and Solaron - Desiccant Heating and Cooling Systems", and is included with this report as Attachment II.

4.1.3 In early August Solaron attended the ERDA Contractors Meeting in Reston, Virginia, and made a presentation of Solaron's work in the development of a Solar/Desiccant Heating and Cooling System. The handouts prepared for this meeting are included with this report as Attachment III.

4.1.4 It was originally planned to do the sub-scale and prototype desiccant system testing in the Solaron laboratory in Denver. To implement the laboratory facilities for this purpose it was necessary to install a fresh air supply and exhaust system. This installation was completed in July. It was later decided, however, that Bry-Air Inc., Sunbury, Ohio had better facilities and capability for doing this work, and could accomplish it at a lower cost to the program.

Solaron prepared a Test Plan in September to cover both sub-scale and prototype system testing. This was sent to Bry-Air with copy to NASA, and, as previously stated, is included with this report as Attachment I.

Prior to this, Solaron had issued Purchase Order No. 1984 to Bry-Air for sub-scale testing of the Model A20A dehumidifier unit purchased for the Akron Site. This testing was scheduled for completion by 10 October 1977. The actual completion date was 2 November and results were received by Solaron, 8 November 1977. Evaluation of these test results will be covered in subsequent reporting.

The tentative decision made at PDR 11/9/77 to use the Heat Pump system in both of the demonstration sites throws a new light on the additional testing to be done on the Desiccant system. NASA did, however, express an interest in completing the test program to compare the actual hardware test results with the computer simulations. Solaron stands ready to implement any further desiccant system testing that NASA desires.

4.2 Desiccant Systems - University of Wisconsin Subcontract.

4.2.1 Based on results of the University of Wisconsin computer simulations which were reported under the title "Simulation of four Open Cycle Desiccant Cooling Systems" it was decided at the Technical Review/Quarterly Review Meeting (TR/QRM) held at Solaron, Denver on 27 June 1977, that the optimum system cycle for the Akron, Ohio site was the Recirculation Cycle, Option 1. This report is included as the Appendix to Attachment II of this report.

As a result of this decision Solaron instructed the University of Wisconsin to simulate the recirculation cycle desiccant system for the model house in New York for one entire year. In this simulation heat exchanger, evaporative cooler and desiccant wheel efficiencies were input to be the same as those available from the equipment planned for the Akron house. The resulting data was used to compare with the simulation data from Carrier on the Solaron heat pump systems combinations. Comparisons were made on total electrical energy consumed by both systems, including fan power and auxiliary heating power for the heating portion of the system, the cooling portion and the total consumed for heating, cooling and hot water.

4.2.2 During August the University of Wisconsin refined the modeling of the recirculation cycle to include fan power consumption and slightly lower heat exchanger performances, before running the whole year simulation of the recirculation desiccant system. The time span to complete the runs and report results was estimated to be 6 to 8 weeks.

4.2.3 In October Solaron received a complete set of New York based simulations of the recirculation desiccant systems at sensible heat exchanger effectivities of 85 and 90 percent. The simulations also included monitoring of the run time of the air conditioning blowers in order to evaluate the amount of parasytic power consumed by the system. These simulation data were then compared to the simulation data generated by Carrier in order to access the energy savings of the two systems. The results of these simulations are presented in Attachment II to this report, along with comparisons to Heat Pump simulations.

4.3 Heat Pump Systems - Carrier Subcontract.

4.3.1 In July, Carrier worked out the modelling errors in the heat pump simulation programs and estimated that results of the runs could be reported within a few weeks.

4.3.2 Results of the simulations were, in fact, received by Solaron in August. The simulations included runs of the three basic system combinations of Solaron's heating system and the Carrier heat pump. Schematics of the three systems, the parallel system, the dual source system, and the off-peak system are given in Figures 1, 2 and 3, respectively. A modeling error was discovered in the off-peak system simulations, so consequently these simulations had to be rerun. Data on the parallel and the dual source system are believed to be valid.

Work continued to correct the modelling of the Solaron/Carrier heat pump systems using the "off-peak" system with storage for cooling. It was also decided to run a second set of "dual source" system simulations using a lower pebble bed temperature limit of 2°C. This simulation run allowed the pebble bed temperature to drop as low as 2°C before the heat pump switched to all outdoor air as a heat source. (In the previous simulations the lowest bed temperature was limited to 15°C.)

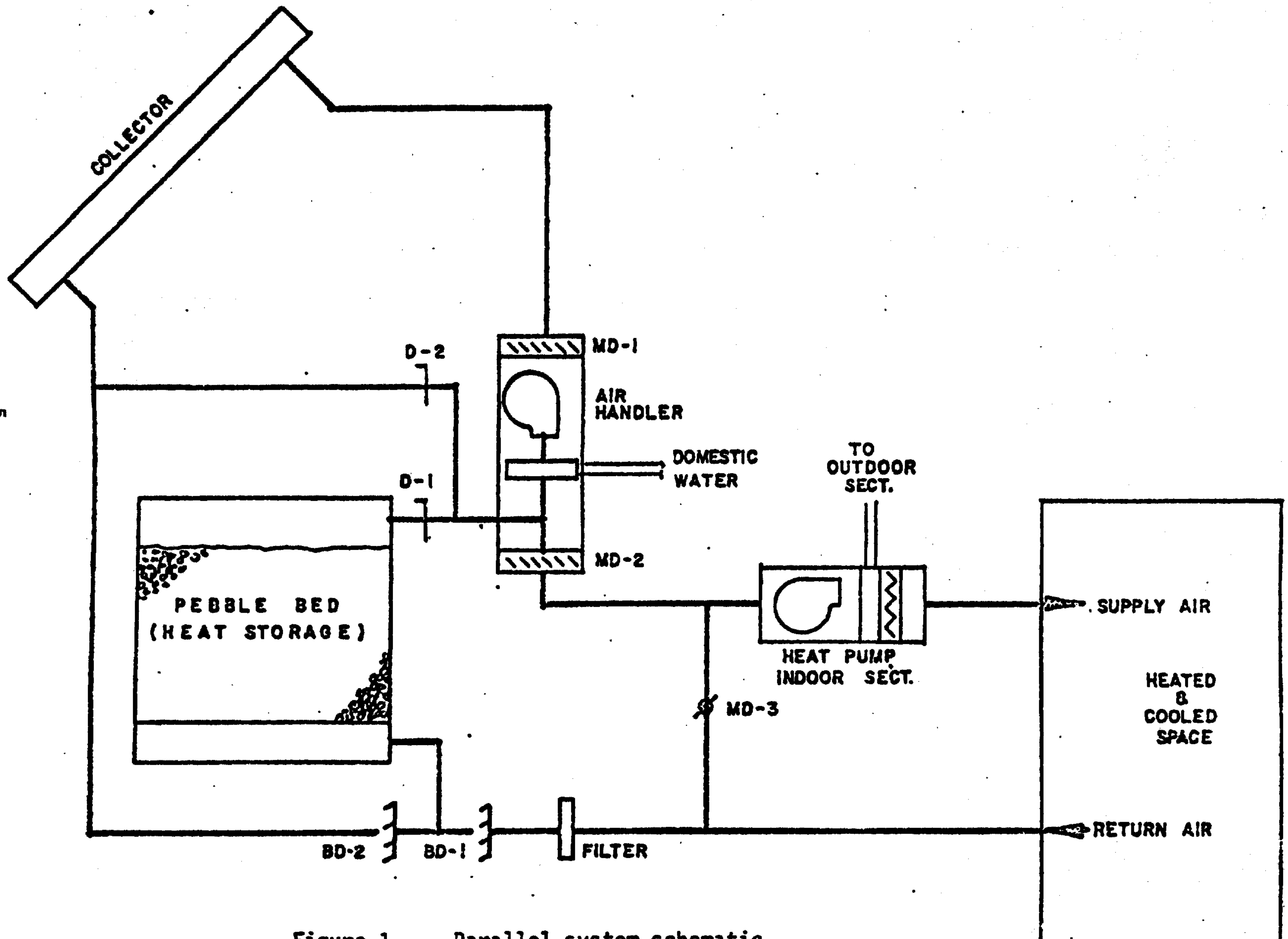



Figure 1 - Parallel system schematic.

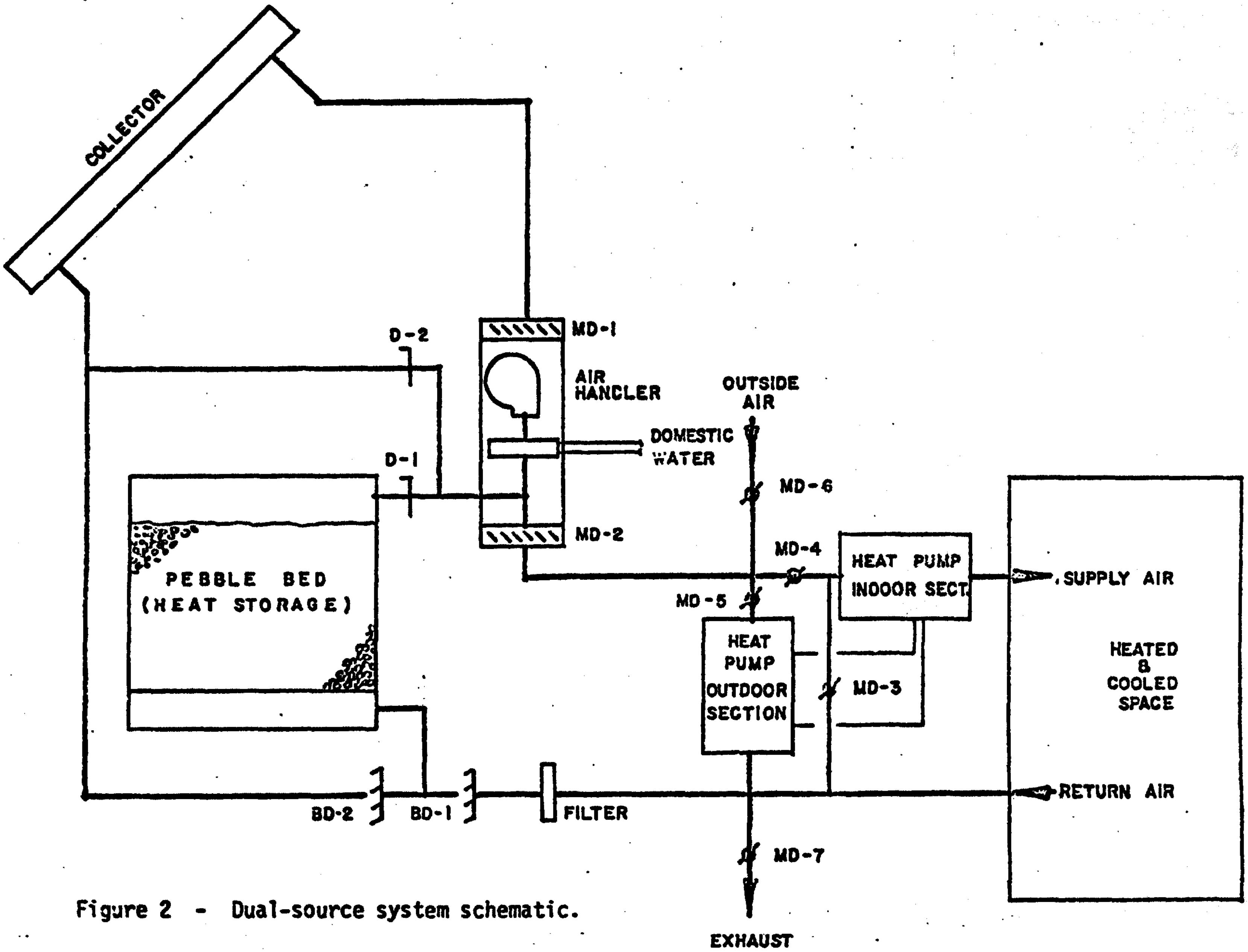



Figure 2 - Dual-source system schematic.

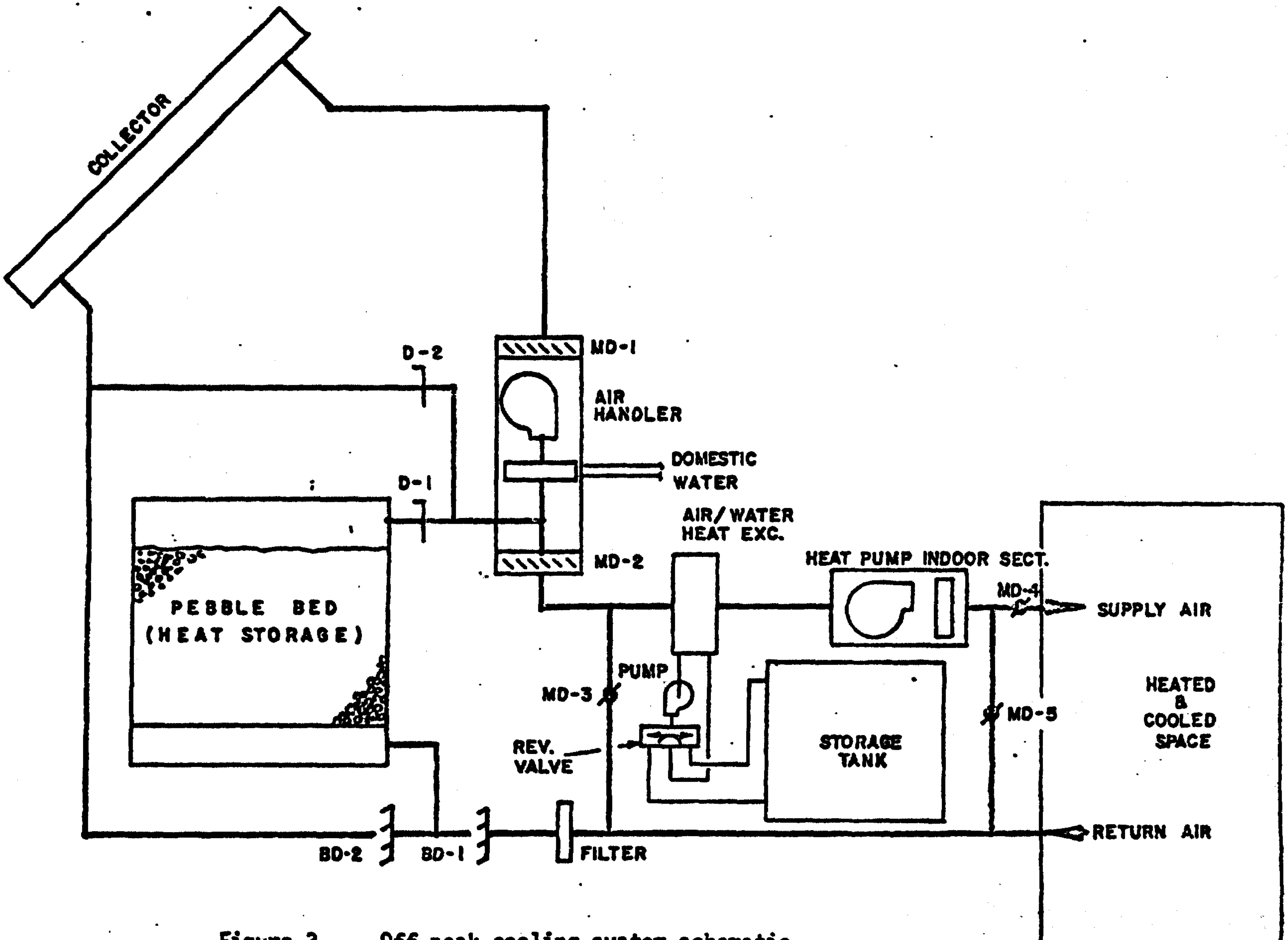



Figure 3 - Off-peak cooling system schematic.

4.3.3 Final simulation results were all received by Solaron in October. A complete analysis of these, along with comparisons to Desiccant system simulations, is presented in Attachment II under the title: "Comparisons of Proposed Solaron - Heat Pump and Solaron - Desiccant Heating and Cooling Systems".

4.4 Installation Planning - Akron House.

4.4.1 An installation planning meeting was held on 20 July 1977 in Akron, Ohio. Attendees representing the various concerned organizations were as follows:

| | |
|---|---|
| R.W. Eisman | Akron Metropolitan Housing Authority |
| Russell F. Barbitta | Architect - Barbitta/James & Assoc. |
| Mitchell Cash | NASA |
| Ralph W. Murphy | NASA |
| Les Shaw | Solaron |
| John Meeker | Solaron |
| Rodney Oonk | Solaron |

At this meeting Solaron presented its background in solar space and hot water heating, installation procedures, and the Solaron Application Engineering Manual. The design and function of the Solar/Desiccant heating/cooling system to be installed in the Akron house were also discussed.

All design and construction details are based on the information contained in the Akron House Data Sheet included with this report as Attachment IV.

4.4.2 During the month of August Solaron prepared preliminary layout drawings of the equipment to be installed in the first demonstration house in Akron, Ohio. The major equipment components are as follows:

(1) Bry-Air desiccant dryer, Model A-20-A as built by Bry-Air, Inc., Sunbury, Ohio.

(1) Temp-X air-to-air plate type heat exchanger, Model A-28-10 as manufactured by Allied Air Products Co., Inc., Newberg, Oregon.

(2) Evaporative Coolers, Model 15 "CC" with Munters Cellulose Media as manufactured by Mountain States Equipment Company, Denver, Colorado.

(28) Solaron Collectors, Model 2001 as supplied by Solaron Corp., Denver, Colorado. (Total of 546 $ft.^2$ collector area).

(1) Rock Box heat storage unit to be built on site as designed and specified by Solaron Corporation.

(1) Air handling unit Model AU-400 - Solaron.

(1) Integrated Control system to control the combined heating and cooling systems in all modes of operation - Solaron.

The sequence of processes in the cooling cycle selected for the Akron site (Recirculation Cycle, Option 1), starting with return air from the house, are as follows:

(Refer to Figures 4 and 5 attached as extracted from Attachment III.)

First, the air drawn from the house (state 6) is passed through the process drying side of the desiccant dryer (in this case a rotating bed of silica gel which is divided into two quadrants (a) process drying and (b) regeneration). In this process moisture is removed from the air by adsorption in the desiccant which simultaneously releases heat to the air stream and results in an increase in the temperature of the air. The enthalpy of the leaving air is also increased.

Air at state (7) is delivered to the hot air side of a plate type air-to-air heat exchanger in which heat is transferred from the process air stream (7)-(8) to the regeneration air stream (2)-(3). The air streams are counterflow.

Air at state (8) is delivered to an evaporative cooler which cools and simultaneously humidifies the air along a constant wet bulb line from state (8) to (9). (9) is the state of the air that must be delivered to the house to maintain design conditions of temperature and humidity in the house. (9)-(6) is the room line. The slope of this line is determined by the sensible heat ratio which, for design purposes, is taken as Total Heat = 1.3 X Sensible Heat (Ref. ASHRAE Handbook of Fundamentals, 1972, Chapter 22, Part III, Page 442). Then Latent Heat = Total Heat - Sensible Heat. The length of this line determines the mass flow of air required to carry the load. Thus at (6), the design state for the house, the process air cycle is completed.

To enable the cycle to operate on a continuous basis, the desiccant must be regenerated. This is done with a separate air stream using 100% outside ambient air.

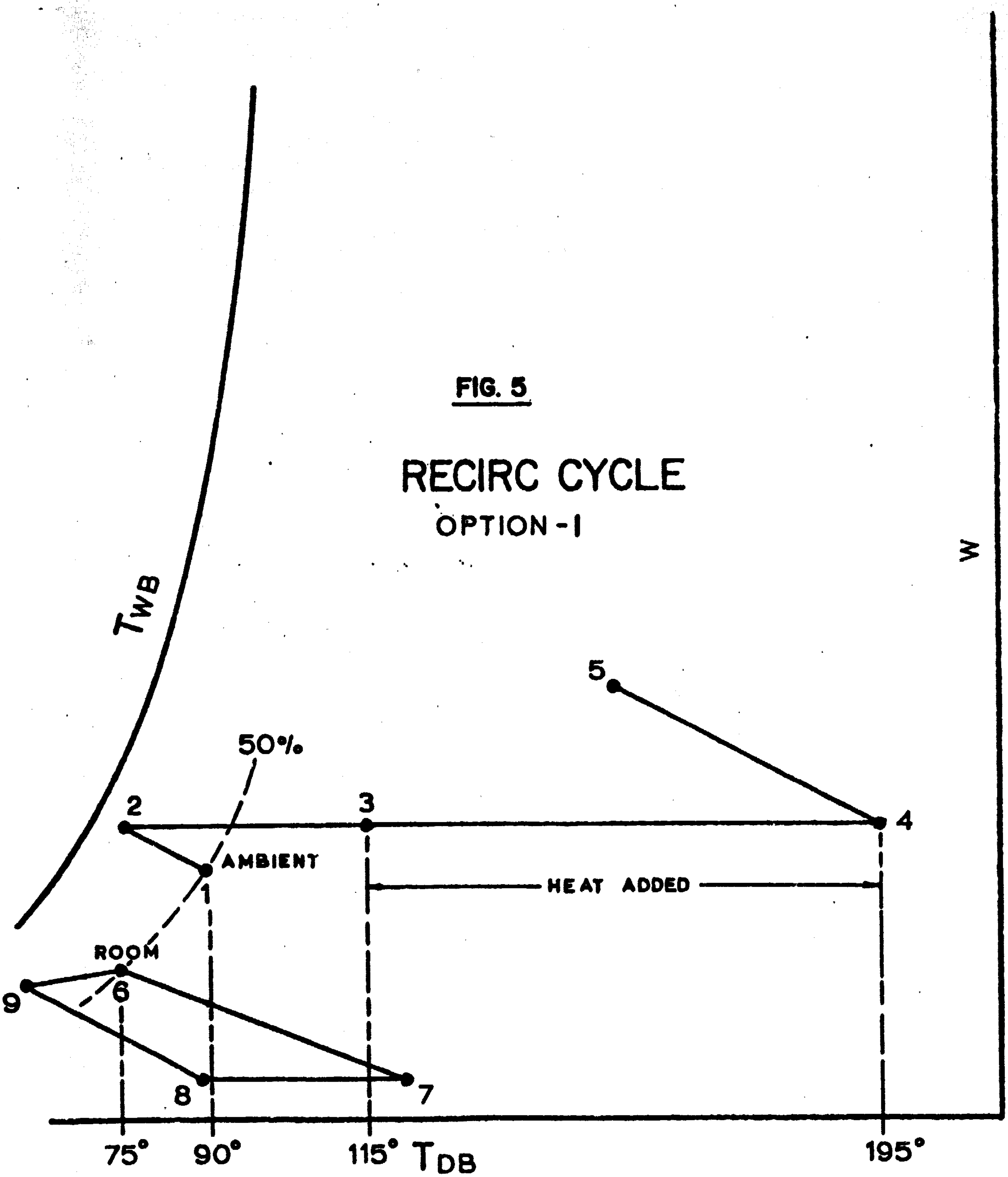
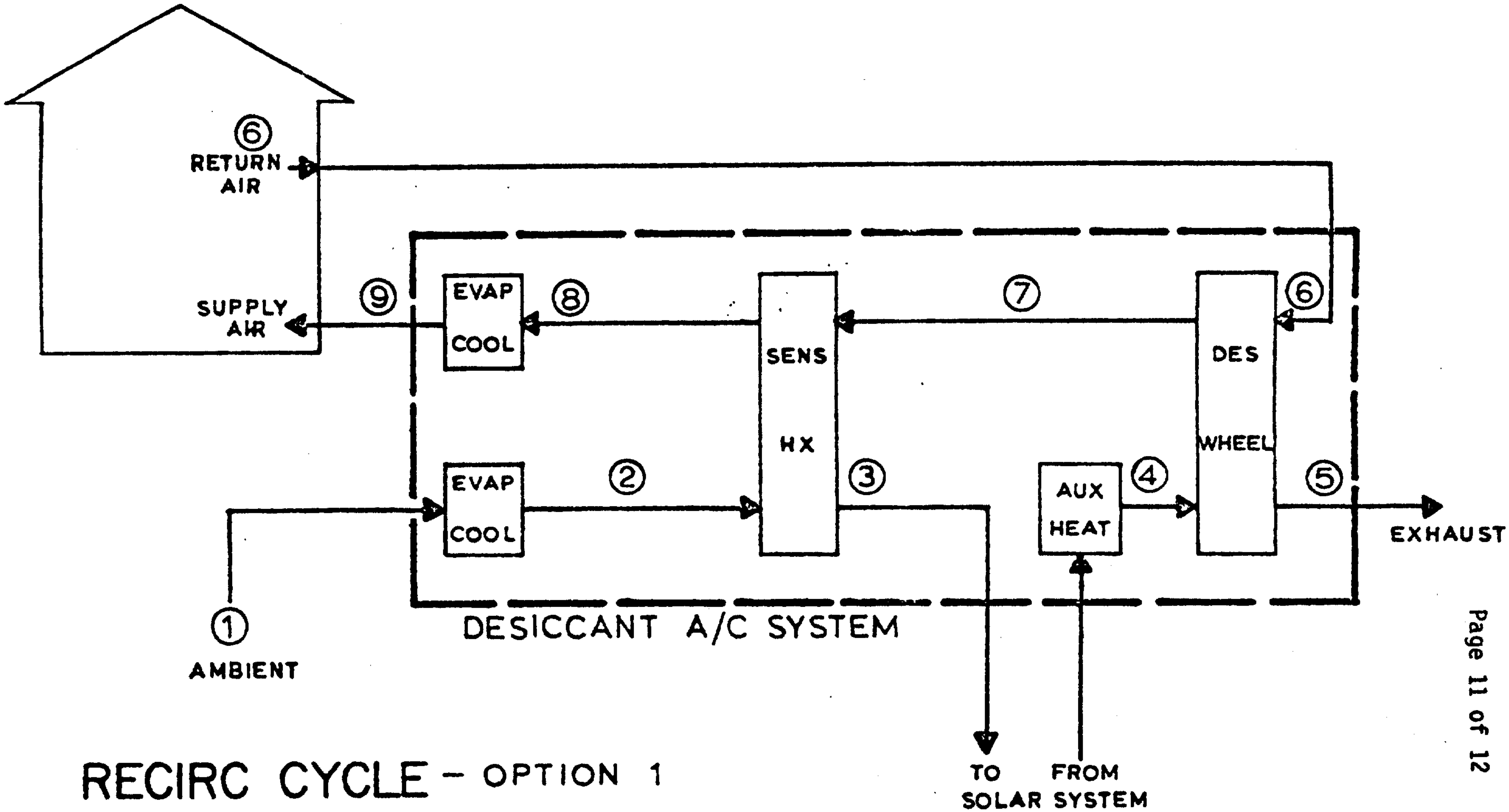


FIG. 5
RECIRC CYCLE
OPTION -1
TWB
W
5
50%
2
3
4
AMBIENT
1
HEAT ADDED
ROOM
9
6
8
7
75°
90°
115°
TDB
195°




RECIRC CYCLE - OPTION 1

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

**FIG. 4**

The ambient air is first cooled and simultaneously humidified from state (1) to (2) using an evaporative cooler similar to the one used for process (8) to (9). This is done to provide a heat sink in the air-to-air heat exchanger for the process air at state (7) and simultaneously to absorb the heat in the regeneration air where it is needed. Design for the Akron site uses a heat exchanger having 85% efficiency. Higher efficiency would be beneficial to overall cycle efficiency, but the sharply rising cost beyond 85% efficiency becomes a controlling factor.

Air at state (3) is then delivered to the solar heating system where it receives additional heat to increase its temperature to that required for regeneration of the desiccant wheel. An auxiliary heater is provided in series to make up any deficiency in temperature left by the solar system.

For this particular system design for the Akron site, it is anticipated that 85 to 90% of the annual summer cooling load would be carried by solar with the remaining 10 to 15% picked up by auxiliary heat. (The system is designed for 65% of the annual winter heating load to be carried by solar). The required entering air state for regeneration of the desiccant wheel is state (4).

The regeneration process is accomplished from state (4) to (5). This completes the regeneration cycle, and the air at state (5) is discharged to atmosphere.

4.4.3 During September the mechanical room layout was refined to streamline the ducting to minimize pressure drop and make the most efficient use of the space available. Design of the house heating/cooling distribution system proceeded along with integration of the solar heating and solar/desiccant cooling systems. The design effort was coordinated with the project architect, Barbitta, James & Associates.

4.4.4 In October the mechanical room layout and details were finalized and trimmed with the emphasis on economics and efficiencies. Design was also completed on the house supply/return ducting and on the supply/return ducting in the attic for the Collectors on the roof. All further work on the desiccant system installation for Akron was terminated on 11/9/77 based on the decision made at the Preliminary Design Review Meeting, same date, that a Heat Pump system would be installed at the Akron site. These drawings will be held in the present state of development pending a possible reactivation of the desiccant system development. As a matter of record for this report, these drawings, D 1000 - House Plans & Elevation, D 1001 - Mechanical Room Plan, Sections & System Schematic, D 1002 - Heat Storage System (Rock Box) are included as Attachment V.

# ATTACHMENTS

I. Test Plan for NASA Contract NAS8-32249.

II. Comparisons of Proposed Solaron - Heat Pump and Solaron - Desiccant Heating & Cooling Systems. (With Appendix).

III. Handouts for ERDA Contractors Meeting - August 8-10, 1977.

IV. Akron House Data Sheet.

V. Akron House Installation Drawings.

D-1000 - Solar Collector Installation and Details.

D-1001 - Mechanical Room Layout - Plan, Sections, Schematic and Sequence of Operations.

D-1002 - Heat Storage Unit (Rock Box).

# ATTACHMENT I



TEST PLAN FOR NASA CONTRACT NAS8-32249

System testing of the Solaron desiccant air-conditioning system will be arranged to yield a maximum amount of information on the operation of individual system components. Input and output air stream conditions as well as water and electrical power inputs will be measured for all system components. The system test schematic will be the standard recirculation cycle Option 1 schematic modified to the extent that the auxiliary heater supplies all heat energy to the system.

For Sub-scale testing all components will be tested in series with each other so that upstream components can serve to provide proper inlet conditions to subsequent components.

In addition to the standard system components a "house load simulator box" will be required to complete the psychrometric cycle, this "box" is to be designed and provided by Bry-Air, subject to approval by Solaron.

Measurement of inlet and outlet conditions of the air will be required for each of the system components. The space and ducting between components will be adequate to provide for good mixing so that accurate measurements of the average air condition can be recorded.

Several sets of conditions must be run to determine performance at "design" and "off-design" loads, to determine system COP's under variable operating conditions, and to determine the ability of the system to maintain desirable, or at least acceptable, conditions in the house. It is also desirable to know the maximum load conditions beyond which the system can no longer maintain acceptable house conditions. These sets of test conditions are described as follows:

<u>Set A</u> - Outdoor summer ambient: 87°FDB, 73°FWB (Akron, Ohio 2½% Summer Design).

Inside Thermostat: 75°FDB,

Inside RH:50%

House load: Sensible: 16,615 Btuh,

latent: 4,985 Btuh,

Total: 21,600 Btuh,

<u>Set B</u> - Ambient: 81°FDB, 69°FWB,

Inside: 75°FWB, 50% RH

House load: Sens: 8,308 Btuh,

lat: 2,493 Btuh

Total: 10,800 Btuh.

(Note: This set is equivalent to 50% of design load on the house.)

Set C - Ambient: 78°FDB, 67°FWB

Inside: 75°FDB, 50% RH

House load: Sens: 4,154 Btuh

latent: 1,246 Btuh

Total: 5,400 Btuh

(Note: This load is equivalent to 25% of design load on the house.)

Set D - Ambient: 89°FDB, 75°FWB
(Akron 1% Outside Summer design)

Inside: 75°FDB, 50%RH (or measure the house conditions maintained if the system cannot maintain the load at design conditions.)

House load: Sens: 19,390 Btuh

latent: 5,817 Btuh

Total: 25,207 Btuh

(Note: This load is equivalent to 116.7% of design load on the house.)

Set E - Ambient: 91°FDB, 78°FWB
(Atlantic City, N.J. 1% outside summer design)

Inside: 75°FDB, 50% RH (or measure the house conditions maintained if the system cannot maintain the load at design conditons.)

House load: Sens: 22,148 Btuh

latent: 6,645 Btuh

Total: 28,793 Btuh

(Note: This load is equivalent to 133.3% of design load on the house.)

Set F - Ambient: 96°FDB, 81°FWB
(Assumed worst case Southeastern U.S.)

Inside: 75°FDB, 50%RH (or measure the house conditions if the system cannot maintain the load at design conditions.)

House load: Sens: 29,076 Btuh

latent: 8,724 Btuh

Total: 27,800 Btuh

(Note: This load is equivalent to 175% of design load on the house.)

The above sets of test to be run (A through F) are summarized for convenience in tabular form attached at Table I.

Since this is a "first-of-a-kind" system and is to be used in a National demonstration program, it is desirable that recorded measurements be as accurate and meaningful as possible consistent with economic and schedule realities.

To meet our contract requirements it is necessary that this test program, including analysis and final report, be completed

Solaron is anxious to work closely with Bry-Air in the planning and execution of this test program. Please call on us at anytime there is a question or need for assistance.

## TABLE I

| Test | Outdoor Design | | Indoor Design (Note 1) | | House Load, BTU/hr. (Note 2) | | | Percent of Full Load (Note 3) | Remarks |
|---|---|---|---|---|---|---|---|---|---|
| | °FDB | °FWB | °FWB | %RH | Sensible | Latent | Total | | |
| A | 87 | 73 | 75 | 50 | 16,615 | 4,985 | 21,600 | 100 | |
| B | 81 | 69 | 75 | 50 | 8,308 | 2,493 | 10,800 | 50 | |
| C | 78 | 67 | 75 | 50 | 4,154 | 1,246 | 5,400 | 25 | |
| D | 89 | 75 | 75 | 50 | 19,390 | 5,817 | 25,207 | 116.7 | |
| E | 91 | 78 | 75 | 50 | 22,148 | 6,645 | 28,793 | 133.3 | |
| F | 96 | 81 | 75 | 50 | 29,076 | 8,724 | 37,800 | 175 | |

Note 1: If for any set the indoor design conditions cannot be maintained, record the best condition of range of conditions that can be maintained.

Note 2: Sensible load X 1.3 = total load per ASHRAE Handbook of Fundamentals, 1972, Chapter 22, Part III, pages 440-443. Then latent load = Total - Sensible load.

Note 3: Full (100%) load is based on Akron, Ohio 2½% Summer Design/ASHRAE. Test Set A is based on this condition.

ATTACHMENT II

COMPARISONS OF PROPOSED SOLARON - HEAT PUMP AND SOLARON - DESICCANT HEATING AND COOLING SYSTEMS

Work funded by NASA Contract No. NAS8-32249

Solaron Corporation
300 Galleria Tower
720 South Colorado Boulevard
Denver, Colorado 80222

October 31, 1977

## I. Heat Pump Simulation

### A. Introduction

Computer simulations of various combined Solaron air solar heating systems and Carrier air-to-air heat pumps were performed in order to access the ability of these systems to save energy for heating, cooling and domestic hot water. The systems analyzed were of three types, (1) A parallel system where the solar system is the primary heating system and the heat pump is used in a conventional fashion (all outdoor air as a heat source) as the auxiliary heat source; (2) A dual source system where the solar system again is the primary heating system but when there is a need for auxiliary heat, the heat pump can extract heat either from the solar system or outdoor air, whichever is warmer; and, (3) An off-peak cooling system, which in the heating mode is the same as the parallel system but during cooling operation has the ability to store cooling in a water tank to be used the next day, thus avoiding usage of on-peak electric power.

### B. System Descriptions

The first system simulated was the parallel system, a schematic of which is shown in Figure 1. Space heating operation of this system is as follows: (a) When solar energy is available and there is a call for heat by the rooms, the system passes air through the solar collectors, through an open MD1 and MD2, through the heat pump indoor unit (blower in unit is on) and into the heated space. Return air from the space passes through the filter and two back draft dampers (BD1, BD2) and back to the inlet of the collector. (b) When the space heating requirement is satisfied or when there is a call for second stage (auxiliary) heat, and solar is available, the solar system will switch to storing heat. Hot air from the collector will be directed through an open MD1 into the top of the pebble bed (MD2 is closed). Cool air from the bottom of the pebble bed passes through BD2 (BD1 is closed) and into the collector.

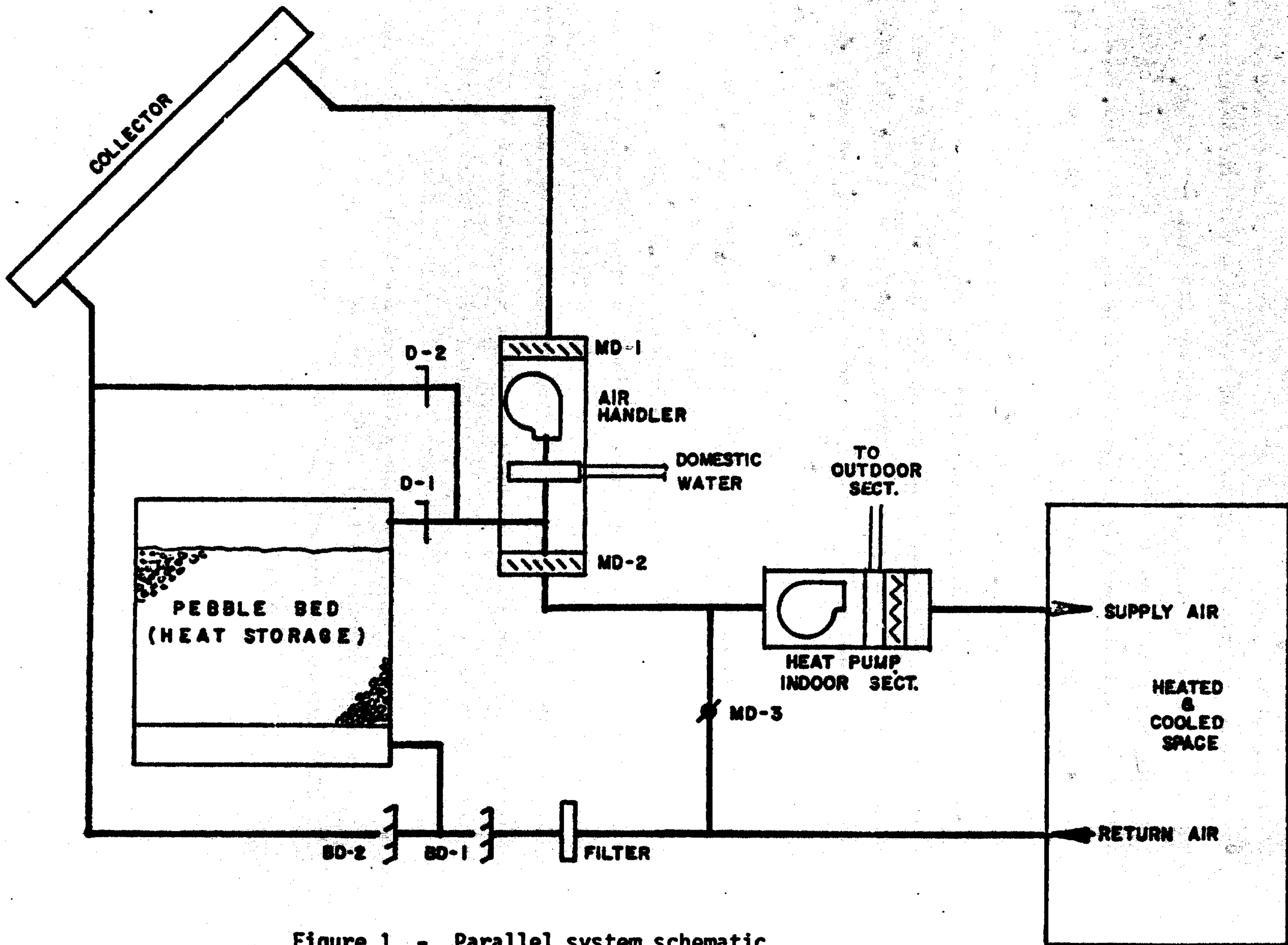



Figure 1 - Parallel system schematic.

In both modes (a) and (b), water is passed through the domestic water coil to extract some of the heat coming from the collector. (c) When no solar is available, and there is a call for first stage heat by the rooms, hot air is drawn from the top of the pebble bed through an open MD2 (MD1 and the solar blower are off), through the heat pump indoor unit, and into the rooms. Cool air from the rooms is passed through the filter and BD1 (BD2 is closed) and returns to the cool end of the pebble bed. (d) When there is no heat left in the pebble bed and no solar is available, or whenever there is a call for second stage heat by the room thermostat, the heat pump compressor comes on and the heat pump supplies heat using outdoor air as the heat source. Under these conditions, MD3 motorized damper opens, MD2 closes and return air is fed directly to the heat pump.

Space cooling (summertime operation) is as follows: (a) When there is a call for cooling, return air is drawn from the rooms through bypass damper MD3 and is cooled by the heat pump. The cool air is then supplied to the house. (b) In order to avoid increasing the cooling load unduly via pebble bed losses by storing heat in the bed all summer, manual dampers D-1 and D-2 are reversed (D-1 is closed and D-2 is opened) so that the pebble bed is not heated and solar heated air is used only to heat domestic water.

Figure 2 is a schematic of the second system simulated, called the <u>dual source system</u>. In this system, the heating modes are more complex and are as follows: (a) When there is solar available and a call for first stage heat, the system will heat directly from the collector. The solar blower will be on, dampers MD1, MD2, MD4, BD-1 and BD-2 will be open (all others closed) and the heat pump blower will be on. (b) If the solar heated air is not warm enough to meet the load directly, then the solar heated air will be fed to the heat pump evaporator section and the heat pump will draw from this air stream. All blowers will be running, dampers MD1, MD2, MD3, MD5, BD-1, BD-2 will be open, with all others closed. If this

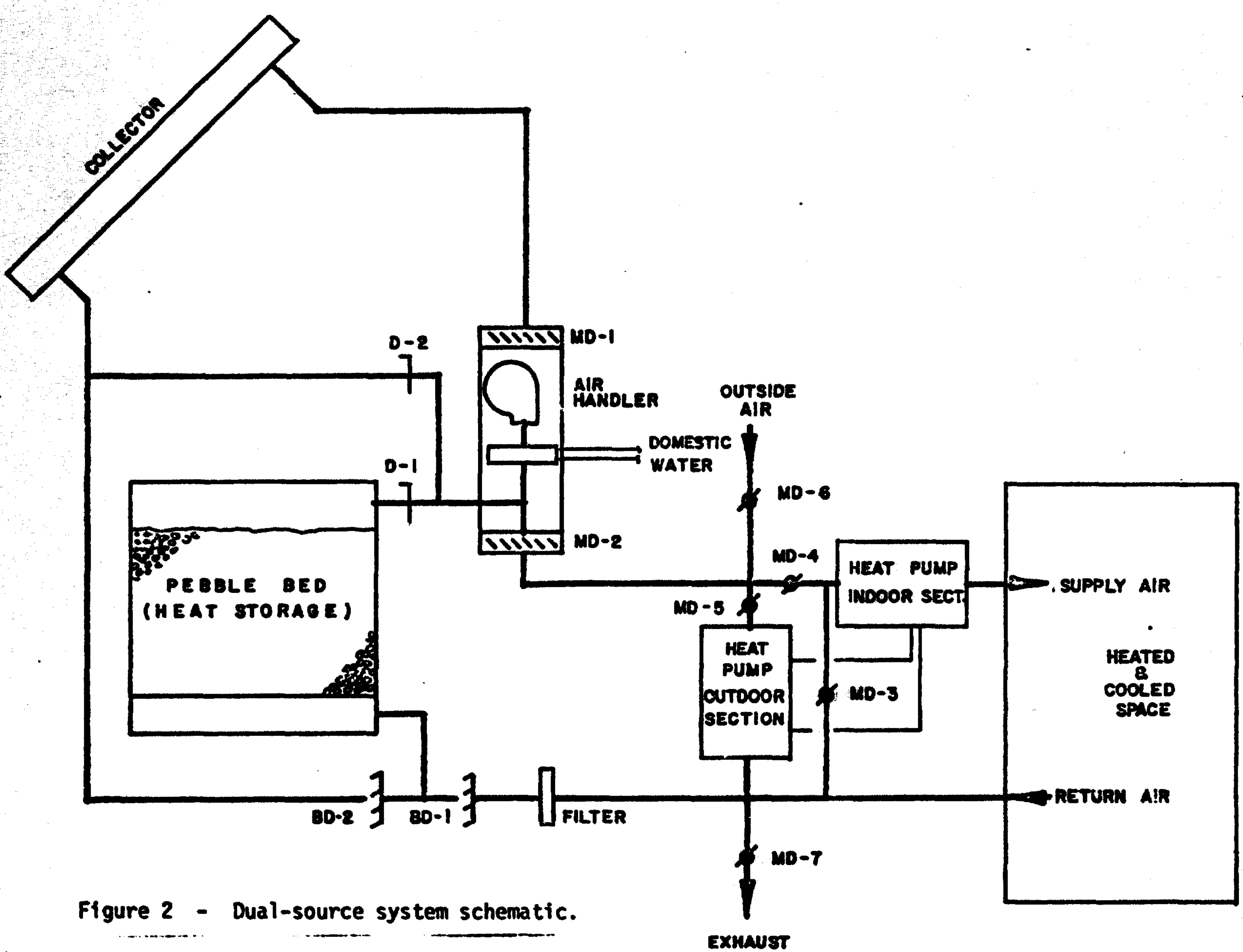



Figure 2 - Dual-source system schematic.

air happens to be too warm for the evaporator's direct use, then dampers MD2, MD6 and MD7 will modulate to mix outdoor air with the solar heated air. (c) When the space is satisfied the system will store heat. System operation will be identical to the operation of the parallel system under these conditions. (d) When heat is required during periods of no solar, the system attempts to heat from storage. The heat pump condenser fan will be on and dampers MD2, MD4 and BD-1 will be open with all others closed. (e) If storage is too cool for direct space heating, the system will switch to solar assisting operation where air from the pebble bed will be supplied to the evaporator section, and the heat pump will supply heat to the load. This type operation will continue as long as there's a call for heat and until either the storage is cooler than outside or until the storage falls to the minimum allowable bed temperature. This temperature in one set of simulations was set at 15°C while in the second set the bed temperature limit was 2°C. Dampers MD2, MD3, MD5 and BD-1 will be open (all others closed) while the heat pump blower and compressor will be operating. (f) When the storage falls to the bed temperature limit or whenever the outdoor air is warmer than storage, the heat pump will use all outdoor air as the heat source. MD3, MD5, MD6 and MD7 will be open. Cooling operation of the dual source system is conventional in that outdoor air is always the heat sink. Whenever there is a call for cooling, MD3, MD5, MD6 and MD7 will open, and the heat pump will extract heat from the rooms and dump it into the outdoor air. As in the parallel system, the solar system will heat domestic water in the pebble bed bypass mode.

The third system that was simulated is a system capable of storing cooling during off-peak power periods. This system is shown schematically in Figure 3. In all heating modes the system operates identically to the parallel system, with dampers MD4 and MD5 being open and closed, respectively. Cooling operation of this system features the off-peak storage feature and is as follows: (a) When

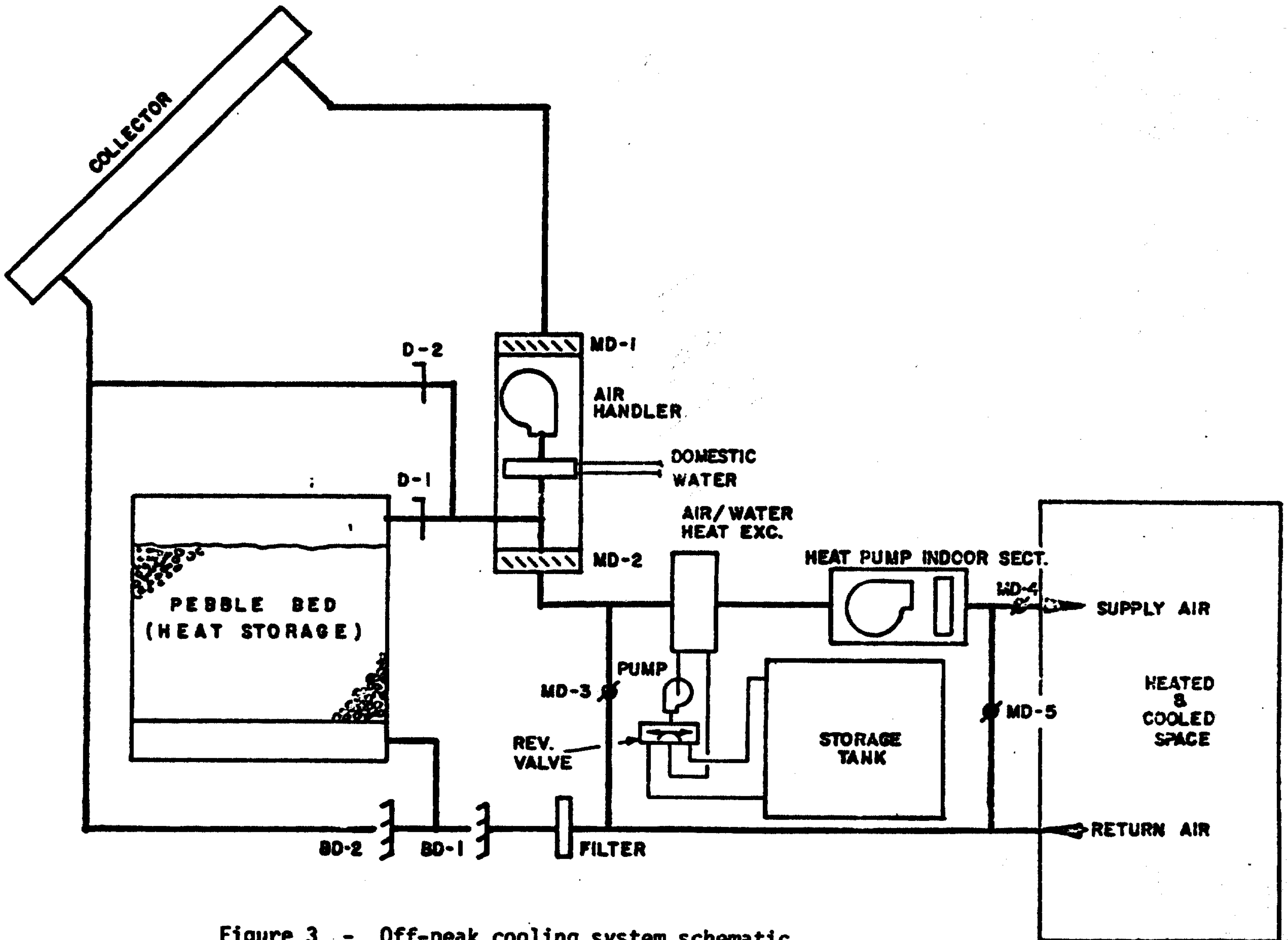



Figure 3 - Off-peak cooling system schematic.

there is a call for first stage cooling, return air passes through MD3 and is cooled by the air/water heat exchanger. This cool air is then fed back to the room by the heat pump fan through MD4. (b) If there is a further call for cooling, the heat pump compressor is energized with air flow the same as above. In both modes the water pump is taking cold water from the bottom of the tank and supplying it to the water loop of the air/water heat exchanger. (c) At night, during presumably inexpensive rate periods, the heat pump is energized to begin the cold storage cycle. Cold air is passed through the room bypass MD5 and through MD3 where it cools the water passing through the air/water heat exchanger. The water passes from the top of the tank through the coil where it is cooled and back to the bottom of the tank. This operation continues until either the tank reaches a low limit of $4^{0}$C or until the off peak rates and (morning).

## C. Simulation Results

The three combined solar heat pump systems previously discussed were modelled on a digital computer using TRNSYS, a solar simulation model developed by the University of Wisconsin. All simulations were performed on the same size residence located in New York, New York. The simulations were run at three collector areas of 29, 58 and 87 square meters. Also runs were made of systems with no solar in order to evaluate the energy savings of the heat pump alone.

One important factor in evaluating system performance is the portion of the total annual heating and domestic water heating load carried directly by the solar system, or the percent solar. The percent solar for all the systems are shown as a function of collector area in Figures 4, 5, 6 and 7. As can be seen from the plots, the systems which do not supply any solar heat to the heat pump evaporator (parallel and off-peak) also supply a greater

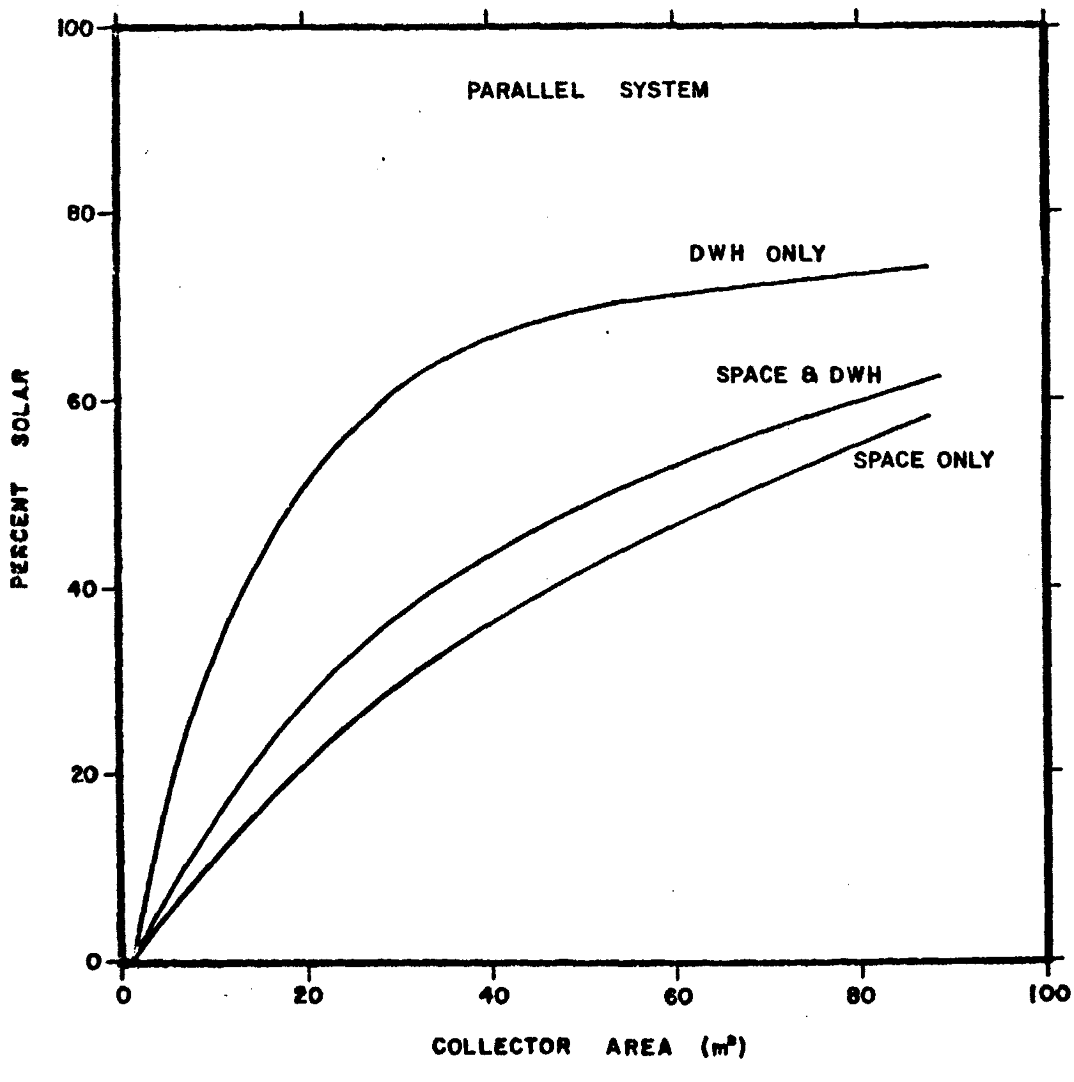



Figure 4 - Percent Solar versus collector area for the parallel system.

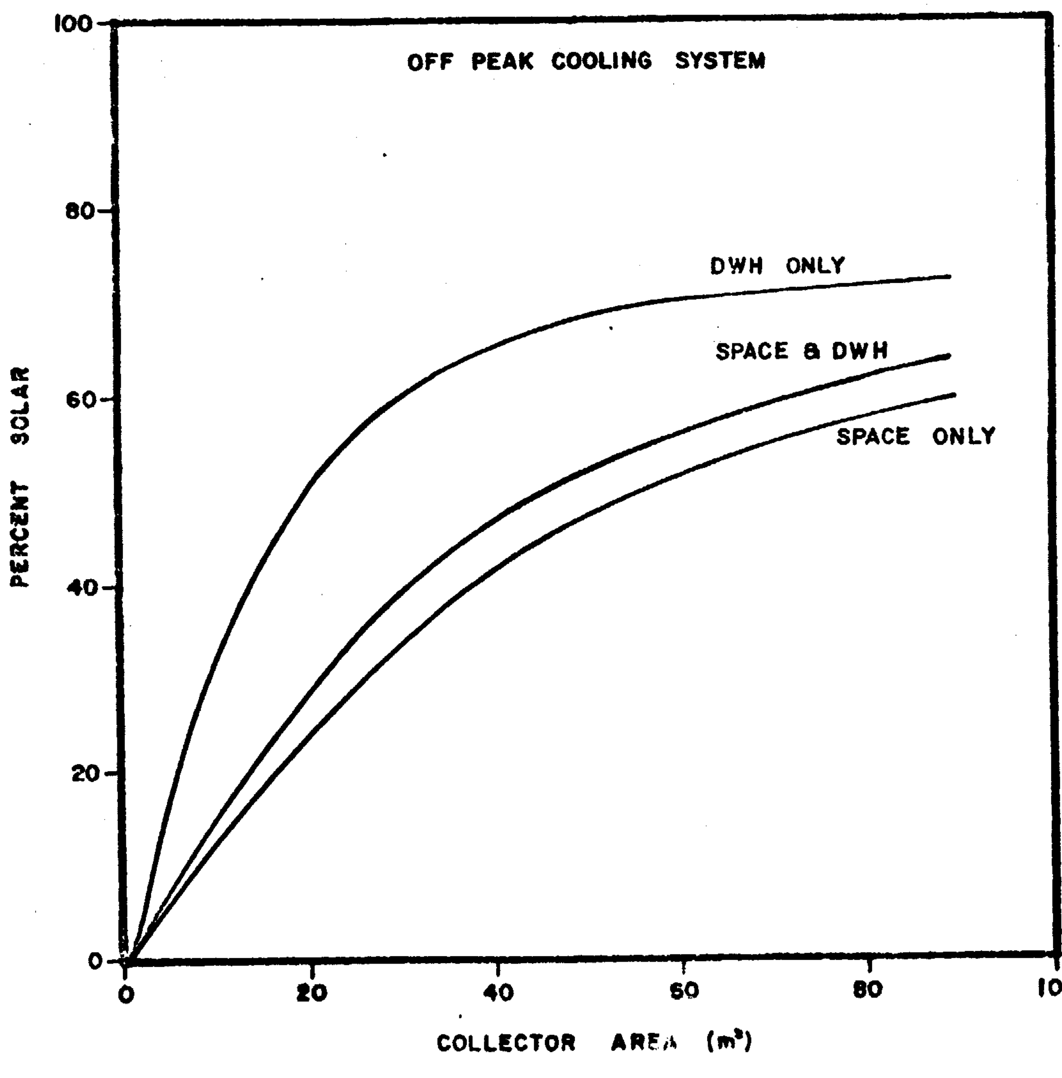



Figure 5 - Percent solar versus collector area for the off-peak cooling system.

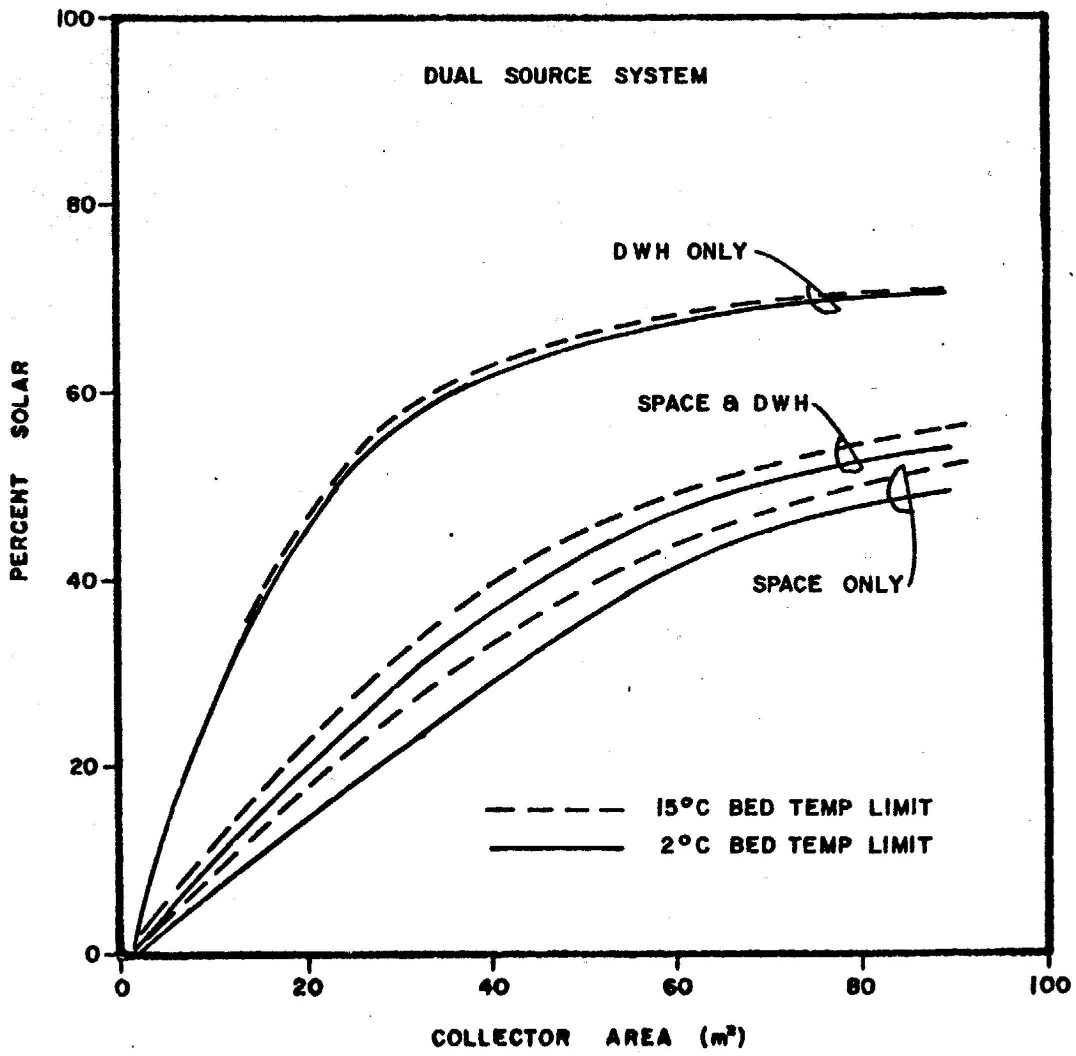



Figure 6 - Percent solar versus collector area for the dual-source system.

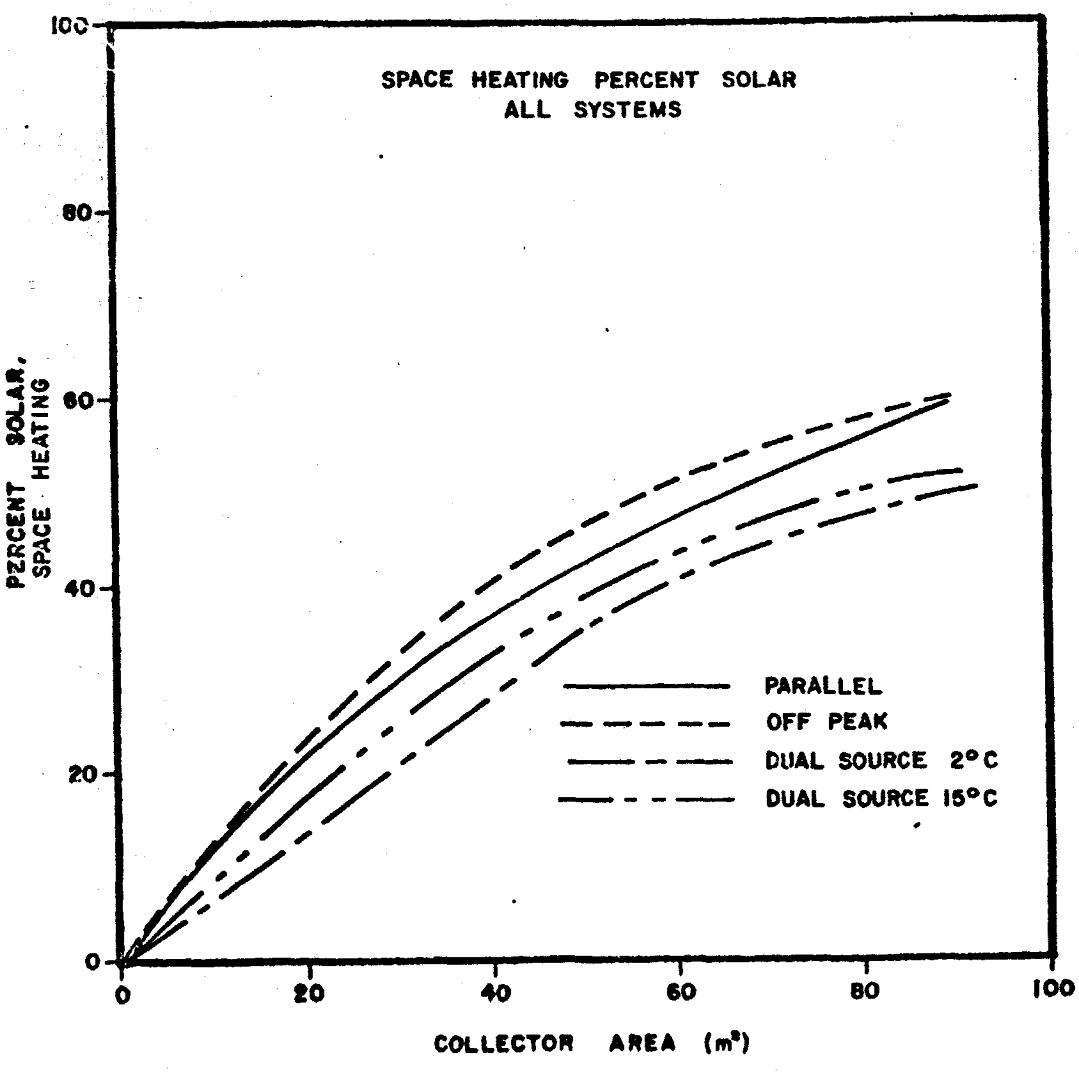



Figure 7 - Space heating percent solar versus collector area for all systems.

portion of the load directly by solar, an expected result. Also the dual source system with a bed temperature limit of 15$^{O}$C supply's slightly more of the load directly by solar than its 2$^{O}$C counterpart for the same reasons.

Figure 8 is a plot of the heat pump coefficient of performance, COP, versus collector area for the various systems studied. As is evident from the plot when examining the COP of the heat pumps at zero collector area (heat pump only), the heat pumps used in the various systems have inherently different COP's, the parallel system heat pump being the more efficient than the dual source system heat pump. This is explained by noting that in the dual source system, since the evaporator section (for heating) is now located indoors and connected to ductwork, rather than outdoors and free standing, a slightly more powerful fan will be needed in the outdoor unit and hence the heat pump will have an inherently lower COP. The heat pump used in the off peak cooling system required a slightly larger indoor fan to overcome the extra resistance due to the extra air/water heat exchanger, and hence that heat pump also had an inherently lower COP.

Examining the slope of the heat pump COP curves in Figure 8 one sees the expected increase in COP of the dual source heat pumps as a function of collector area. Examination of the parallel system heat pump COP shows a slight increase at small collector areas before the gradual decline at large collector areas. This can be explained by noting that for the particular year of New York weather data examined, cold weather was also associated with clear weather and therefore the solar system supplied heat when the heat pump at zero collector area was operating at a low COP. At larger collector areas the solar system carried all the load in the mild months in spring and fall, and thus the heat pumps average COP began to decline due to it running only during the colder months of winter.

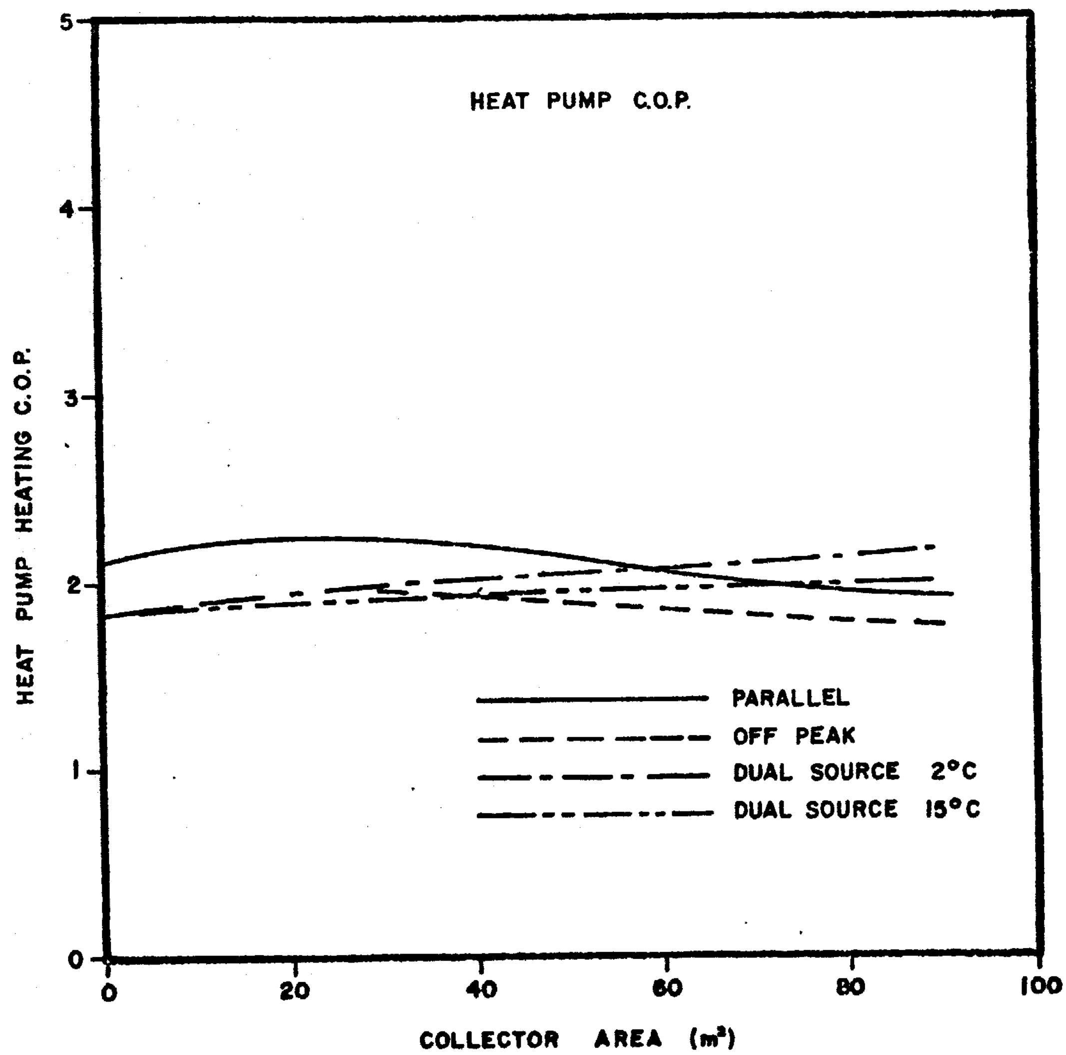



Figure 8 - Heat pump heating COP versus collector area for all systems.

The net effects of higher percent solars for the parallel and off-peak systems coupled with their lower heat pump COP's versus the dual source systems are shown in Figres 9 and 10. Figure 9 shows the electric power consumption by the various systems for space heating. The electric power consumption includes the power to run all the fans and the compressor in all the systems. As can be seen from the plots the parallel system uses the least amount of electric power for space heating at all collector areas. This result is also evident in Figure 10 where the parallel system has the highest system space heating COP.

The annual performance of the various systems in their respective cooling modes are shown in Figures 11 and 12. In Figure 11 the electrical energy consumed for space cooling is shown. Note that the parallel system uses the least power for cooling, due to the heat pump having the highest inherent COP. The off-peak system uses the most power due to the heat pump fan running much more often, first to store the cooling in the water tank and then to extract it from the tank the next day. Also all the space cooling loads increase as a function of collector area, due to greater heat losses from the pebble bed (in the spring and the fall) and the solar domestic water preheat tank (losses which occur the year round).

The results of all the simulations are summarized in Figures 13 - 16. As is evident in comparing the Figures the parallel system consumes the least amount of power for heating, cooling and domestic hot water, and hence has the highest total system COP.

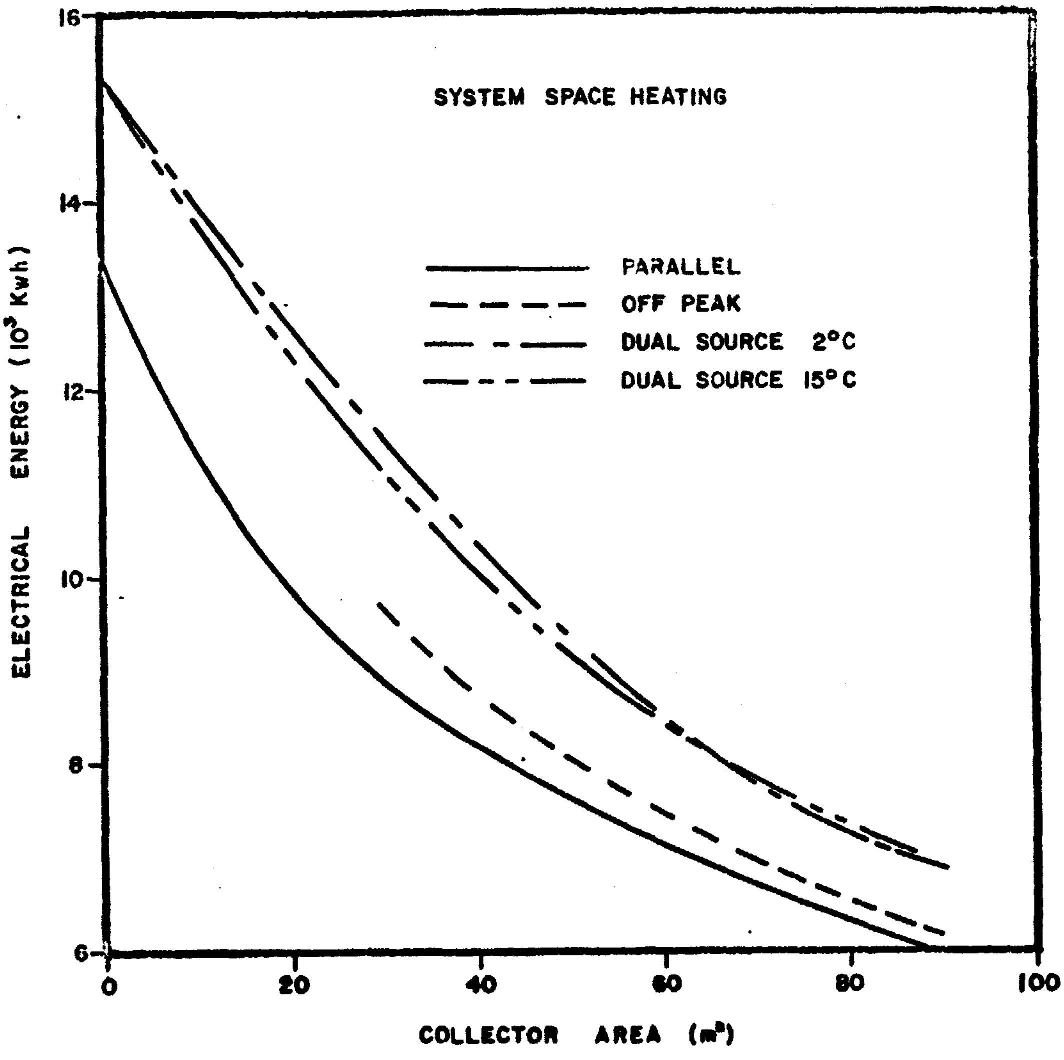



Figure 9 - Electrical energy consumed for space heating versus collector area for all systems.

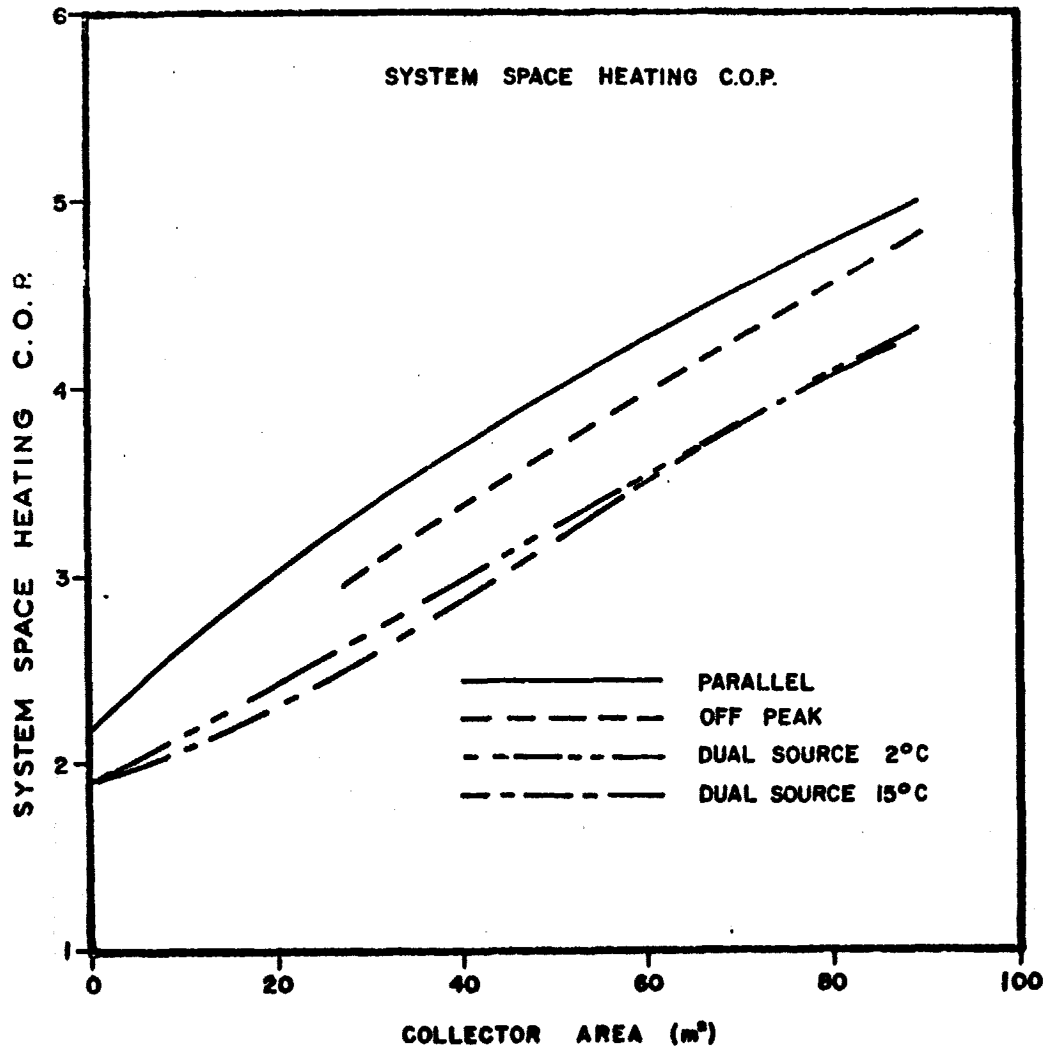



Figure 10 - System space heating COP versus collector area for all systems.

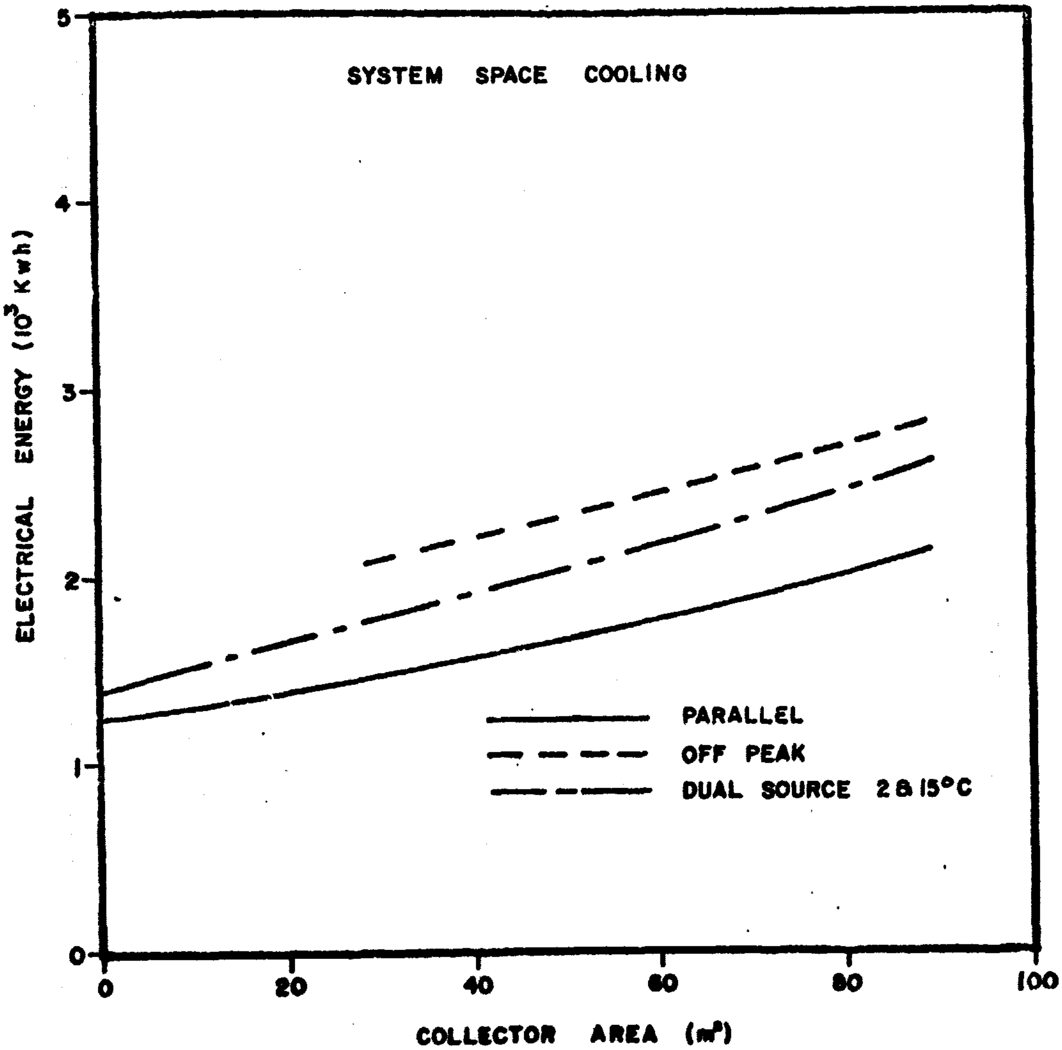



Figure 11 - Electrical energy used for cooling versus collector area for all systems.

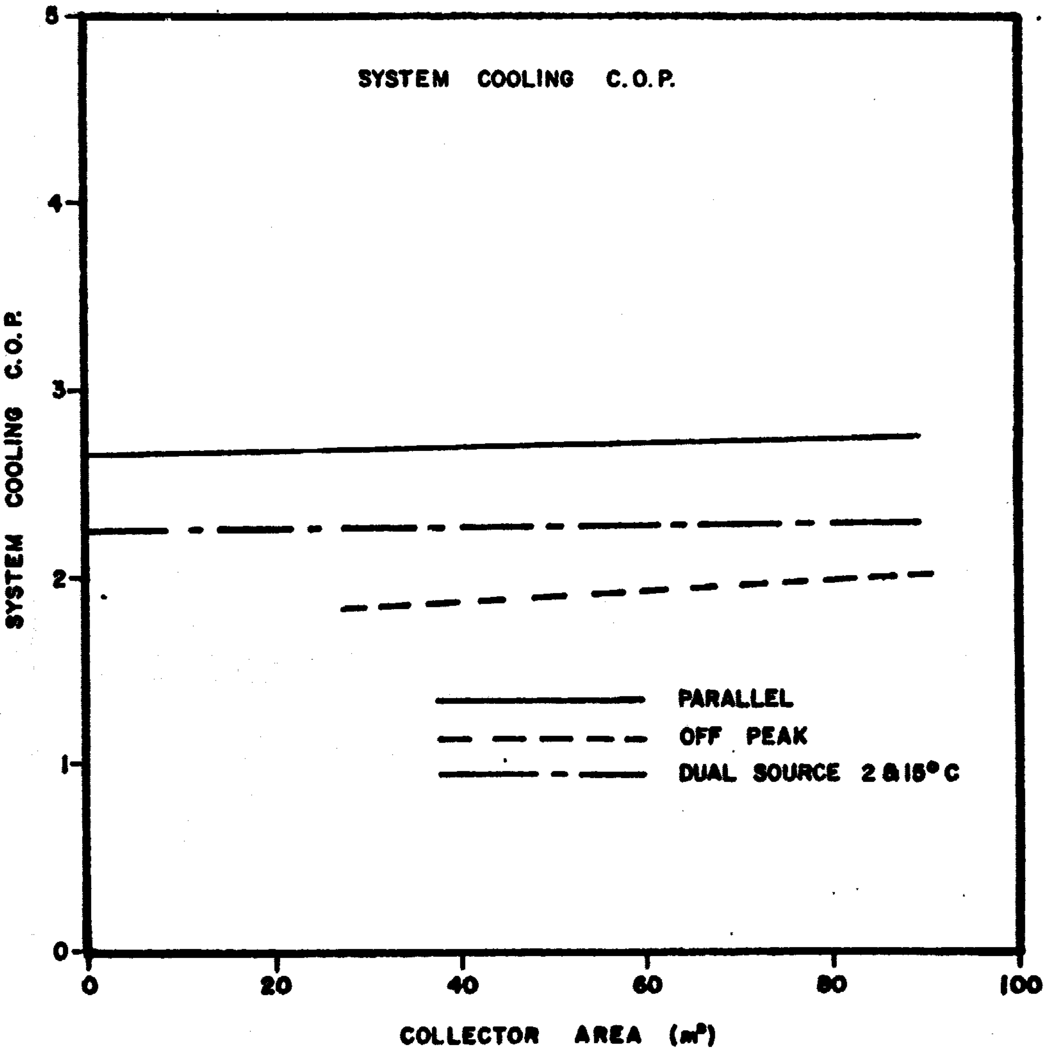



Figure 12 - System cooling COP versus collector area for all systems.

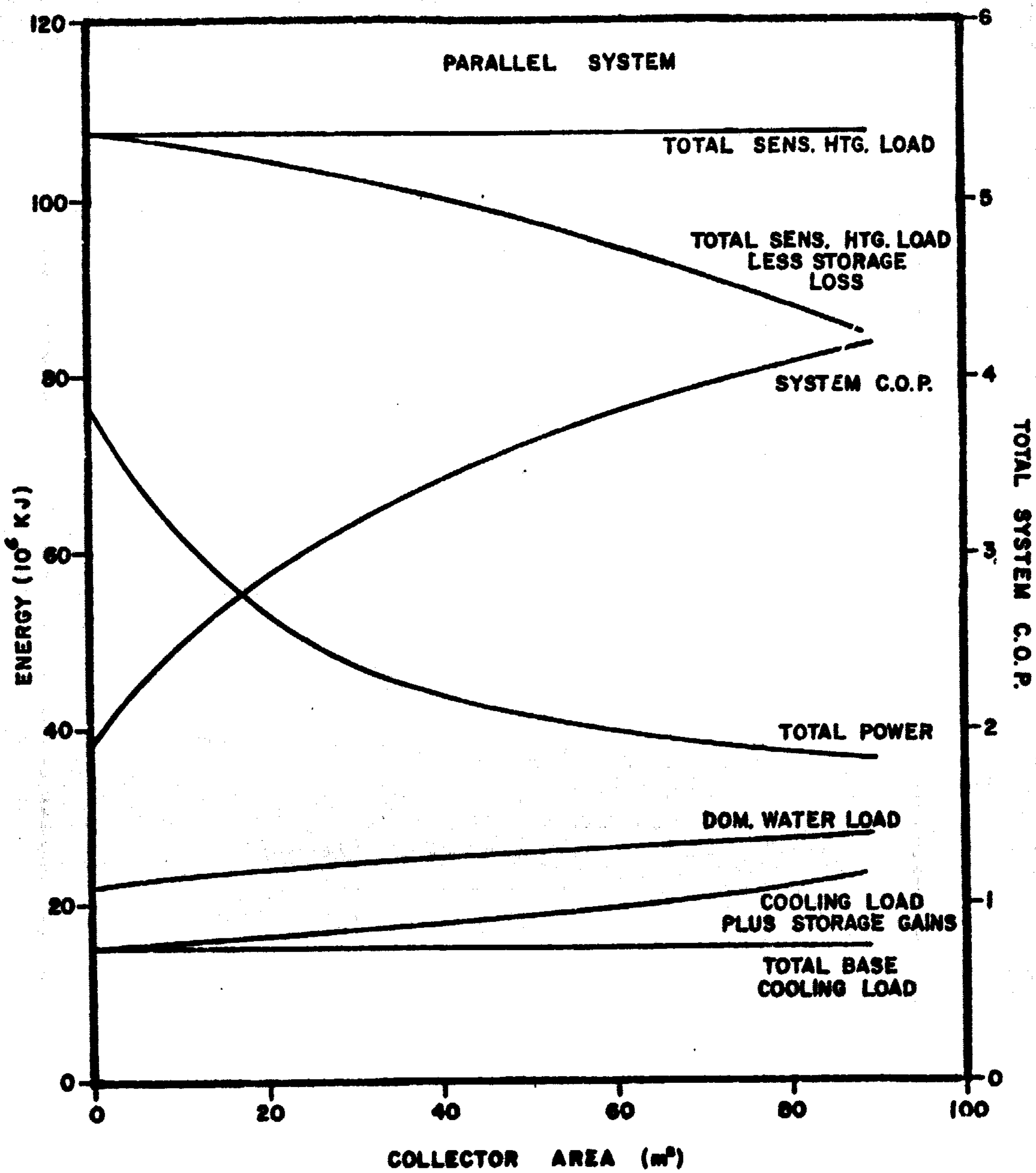



Figure 13 - Pertinent energy quantities and total system COP versus collector area for the parallel system.

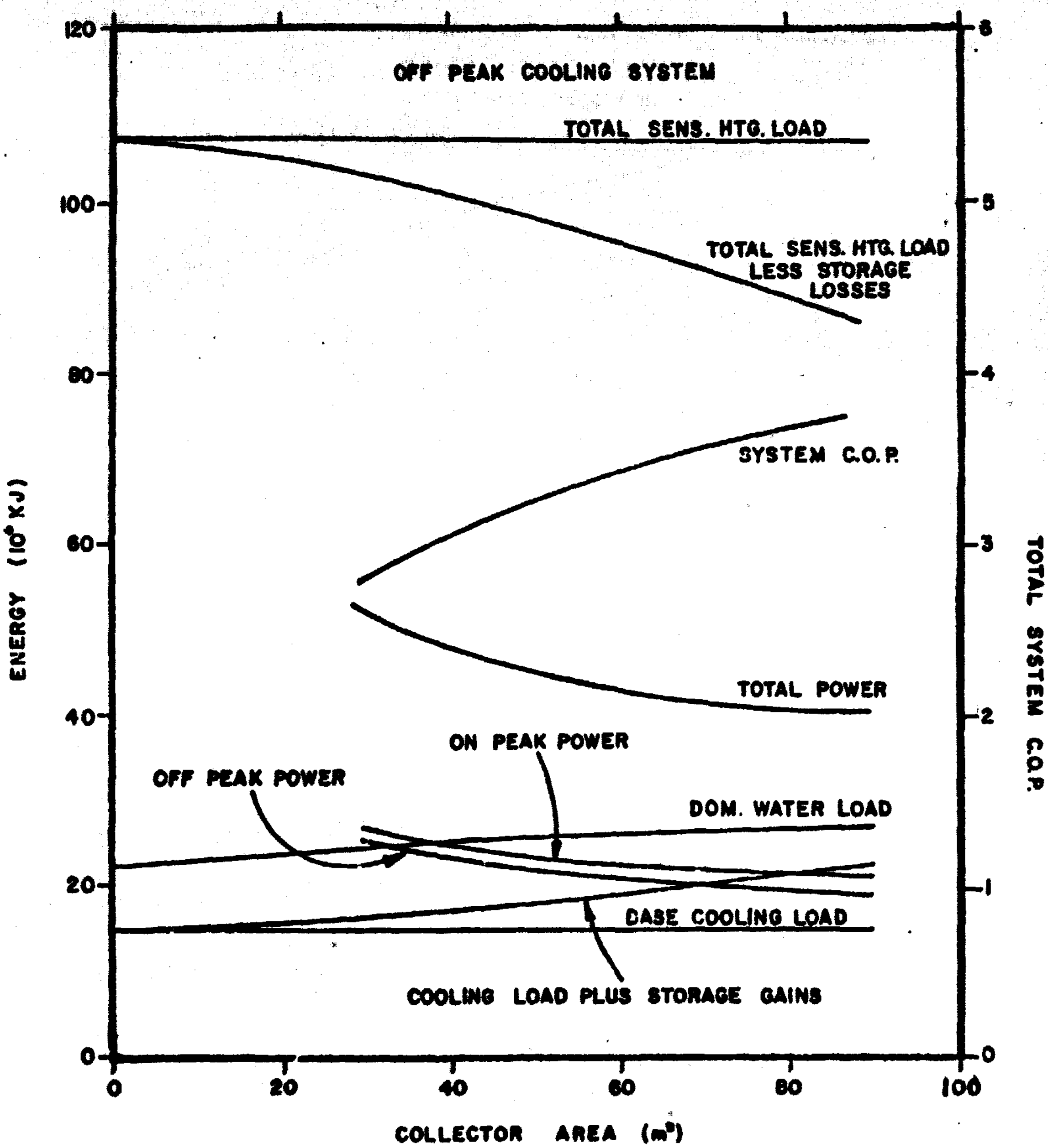



Figure 14 - Pertinent energy quantities and total system COP versus collector area for the off-peak cooling system.

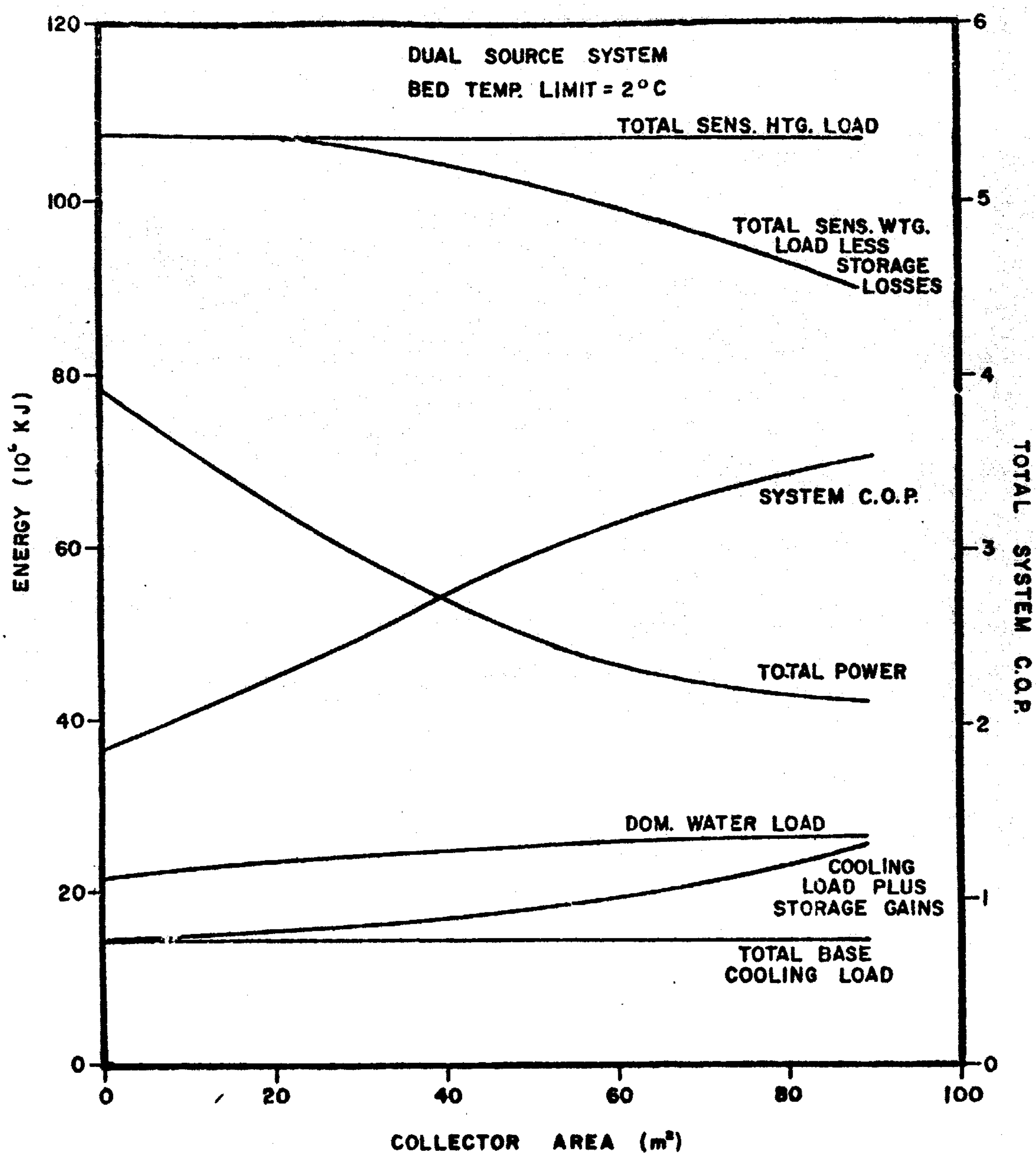



Figure 15 - Pertinent energy quantities and total system COP versus collector area for the dual source system at a bed temp. limit of 2°C.

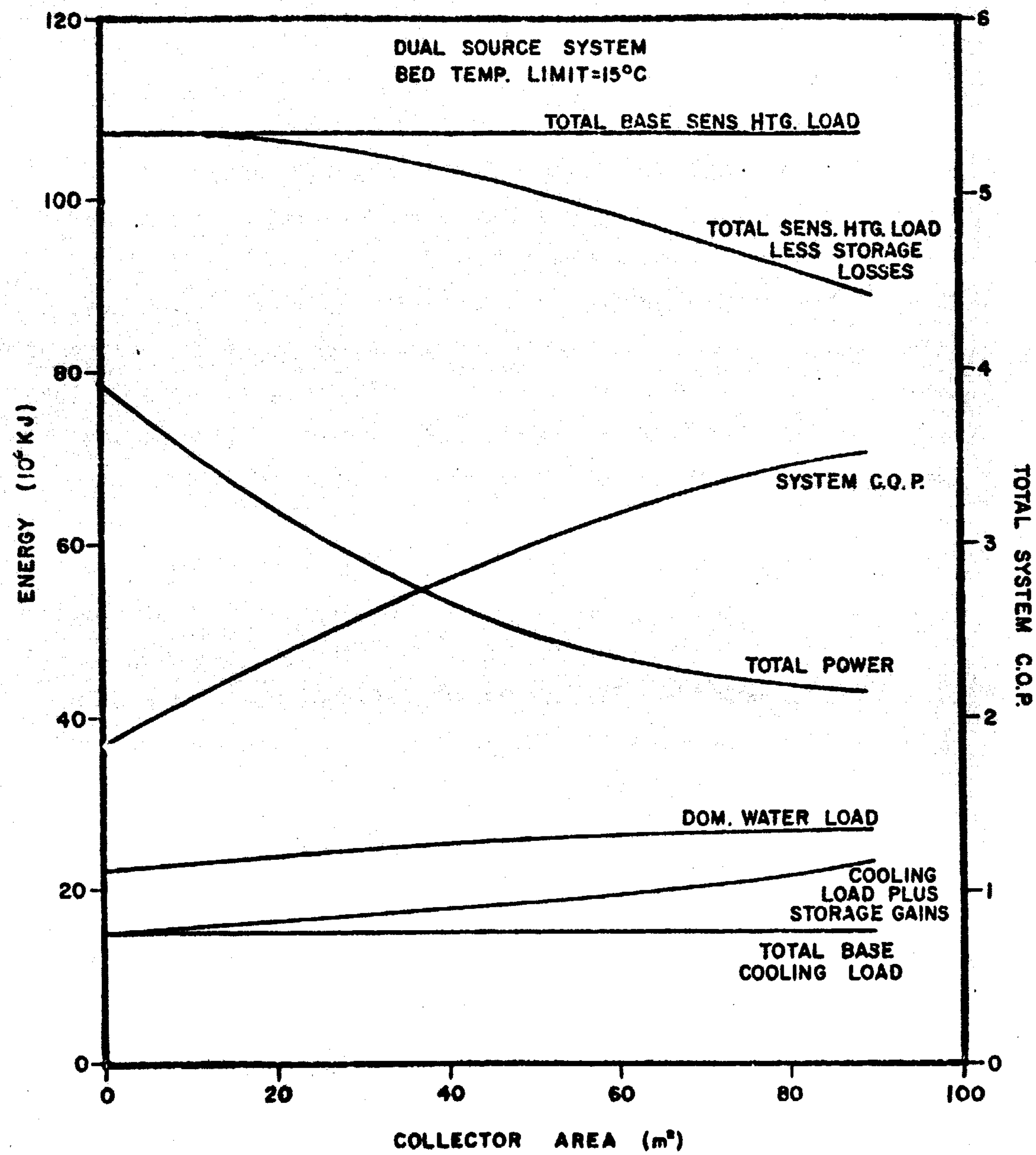



Figure 15 - Pertinent energy quantities and total system COP versus collector area for the dual source system at a bed temp. limit of 15°C.

## II. Desiccant System Simulations

### A. Introduction

Several combined Solaron solar air heating and solar/desiccant cooling systems were analyzed on a digital computer. All these systems use a silica gel desiccant wheel as a means of dehumidification in conjunction with a sensible heat exchanger and evaporative cooler to provide sensible and latent cooling for the house. The desiccant wheel is regenerated by solar and/or auxiliary heat. In all, four variations of solar augmented silica gel desiccant cooling cycles were examined. These cycles include the two basic solar -MEC cycles, the ventilation and recirculation cycle, and two modifications to the recirculation cycle.

The four cycles discussed above were modelled for the cooling season only for the same house as used in the heat pump simulations for the New York location. The results of these simulations are given in the appendix. From these simulations the recirculation cycle desiccant system was chosen as the cycle with the best overall cycle COP and least amount of equipment.

### B. System Operation

Figure 17 shows the desiccant recirculation cycle, option 1, integrated into the Solaron system. The system's heating operation is as follows: (a) When solar is available and there is a call for heat in the rooms, air is circulated through damper MD1-A, MD2-A, and MD6 into the rooms. Return air passes through MD4, BD-1 and BD-2 and back to the collectors. (b) When there is no longer a requirement for heat in the rooms the system will switch to storing heat. In this mode MD2 switches, blocking flow to the rooms and instead diverting flow to the top of storage. Cool air from the bottom of storage passes through BD-2 and returns to the collector. (c) When heat is required and no solar is available, the system

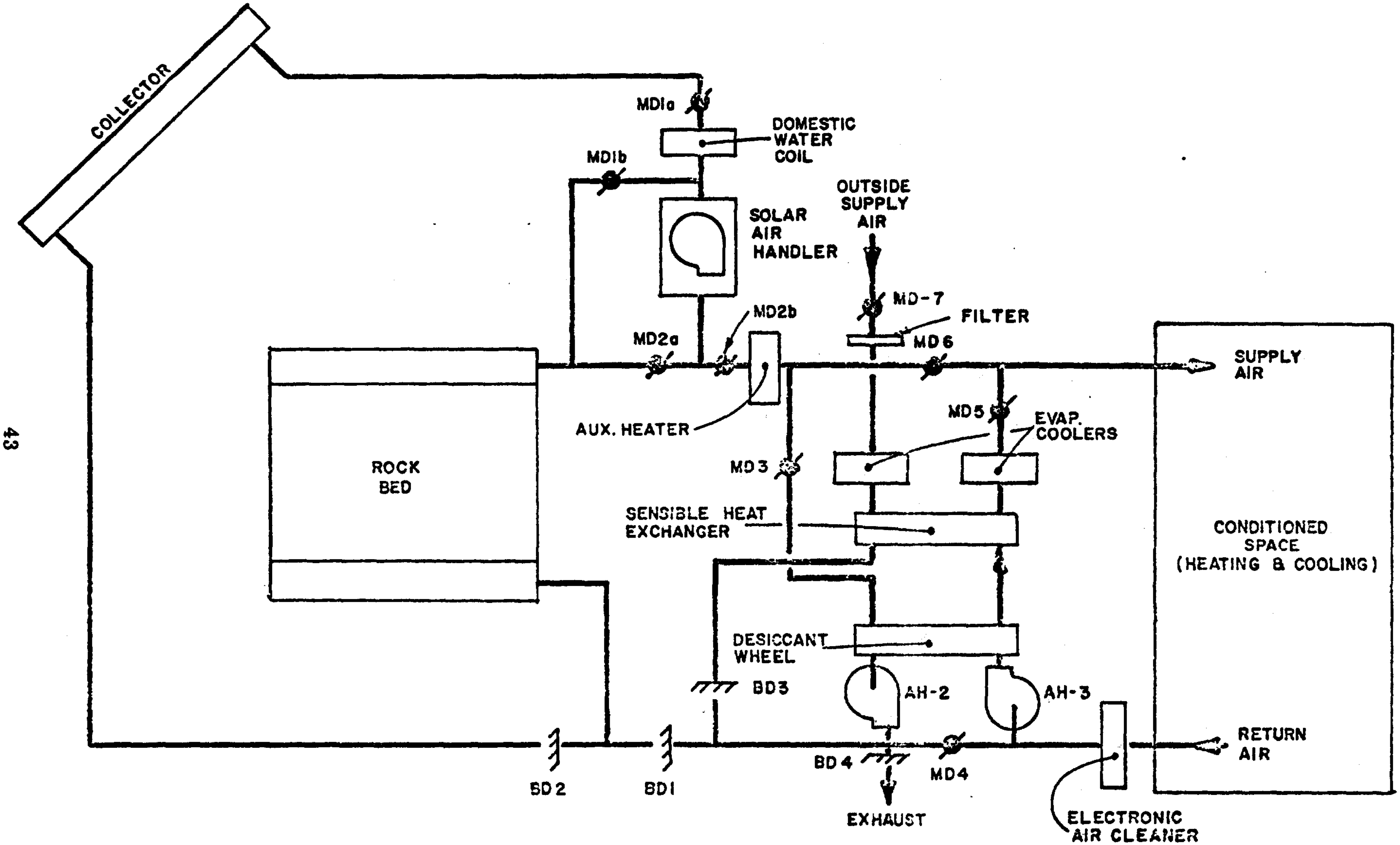



Figure 17 - Recirculation desiccant system schematic.

will heat from storage. In this mode hot air from the top of storage is drawn through MD1-B, MD2-B, MD6, and delivered to the space. Return air flows through MD4, BD-1 and into the bottom of storage. (d) If the room heating requirements are not satisfied, or if the storage is depleted, the system will bring on the auxiliary heater with flow being the same as in the heating from storage mode.

Cooling operation of the Solaron recirculation, Option 1, desiccant system is as follows: (a) If there is a call for cooling by the rooms and solar is available, return air from the rooms is drawn by AH-3 and blown through the desiccant wheel, where it is dried and heated. This hot dry air is then sensibly and evaporatively cooled. This cool air then passes through MD-5 and returns to the house. On the regeneration side, outside air passes through MD-7, through the filter and then to the evaporative cooler. The evaporatively cooled air is then heated by the sensible heat exchanger, passes through BD-3, BD-1 and BD-2 and enters the collector. Hot air from the collector passes through MD1-A, MD2-B and through the auxiliary heater. The auxiliary heater adds heat to the air, if required, after which the air passes through the desiccant wheel, removing moisture from it, and is exhausted. (b) If no solar is available and there is a call for cooling, return air passes through AH-3 as in (a). Regenerative outside air passes through the evaporative cooler and sensible heat exchanger as before, but then enters the cool end of the pebble bed. Hot air from the top of the pebble bed passes through MD1-B, through MD2-B and through the auxiliary heater. The auxiliary heater again adds heat if required, after which the hot air is used to regenerate the desiccant wheel and is exhausted. (c) When solar is available and no cooling is required the system will store heat in the pebble bed. This mode is the same in the summer as it is in the winter.

## C. Simulation Results

The desiccant recirculation cycle systems were modelled using the same house load and New York data as were used in the heat pump modelling. The desiccant system was modelled at three different collector areas of 25, 50 and 75 square meters, and the simulations were performed using TRNSYS.

Figure 18 is a graph of the percent of the space heating and domestic water heating loads carried by solar. Note that the percent solar for space heating, water heating and combined space and domestic water heating of the desiccant system have the same general shape and values as do the parallel heat pump percent solars. However, careful comparison shows them to be about 7 to 10% higher. These differences can be explained by noting that in the heat pump simulations the percent solar was calculated using the heat delivered by the system as the value of the denominator and the portion supplied by solar (collector and storage) in the numerator. In the desiccant simulations, the percent solar was calculated using the total heating load as the denominator and the portion supplied by solar plus the heat storage losses to the load in the numerator. The second ratio is inherently higher than the first.

Regardless of how the percent solar is calculated, the parallel heat pump simulations should supply approximately the same amount of auxiliary heat for space heating as does the desiccant system. Examination of Figure 19 shows this to be true. The parallel heat pump system supplies slightly more auxiliary heat due to this system not continuing to extract heat from the pebble bed when the room requires second stage heating as the desiccant system does.

Figure 20 demonstrates the benefits of using the heat pump as the back-up heating source. As can be seen from the plot, the COP of 2 for the heat pump saves a large amount of electrical energy for heating on an annual basis.

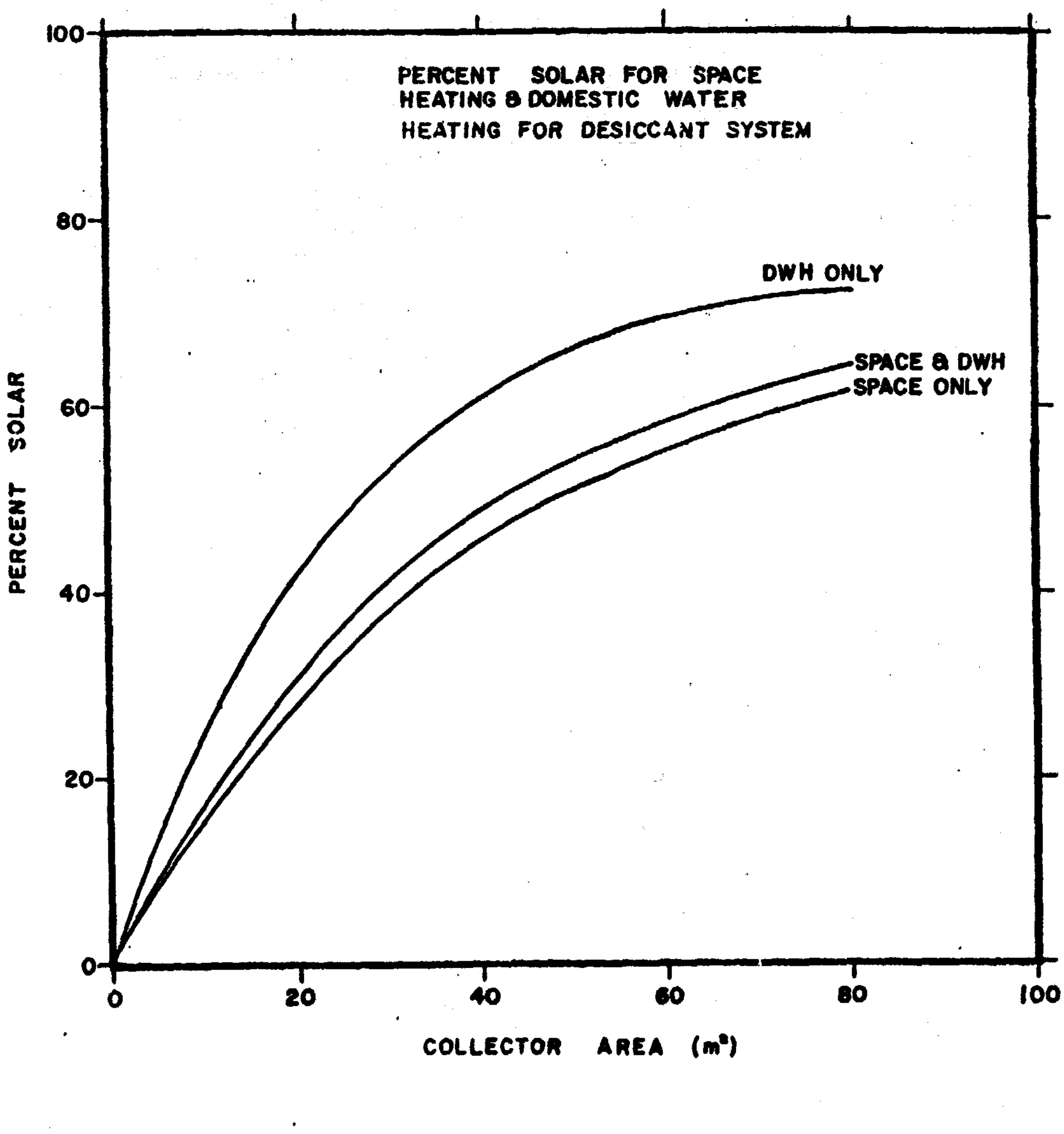



Figure 18 - Percent solar versus collector area for the desiccant system.

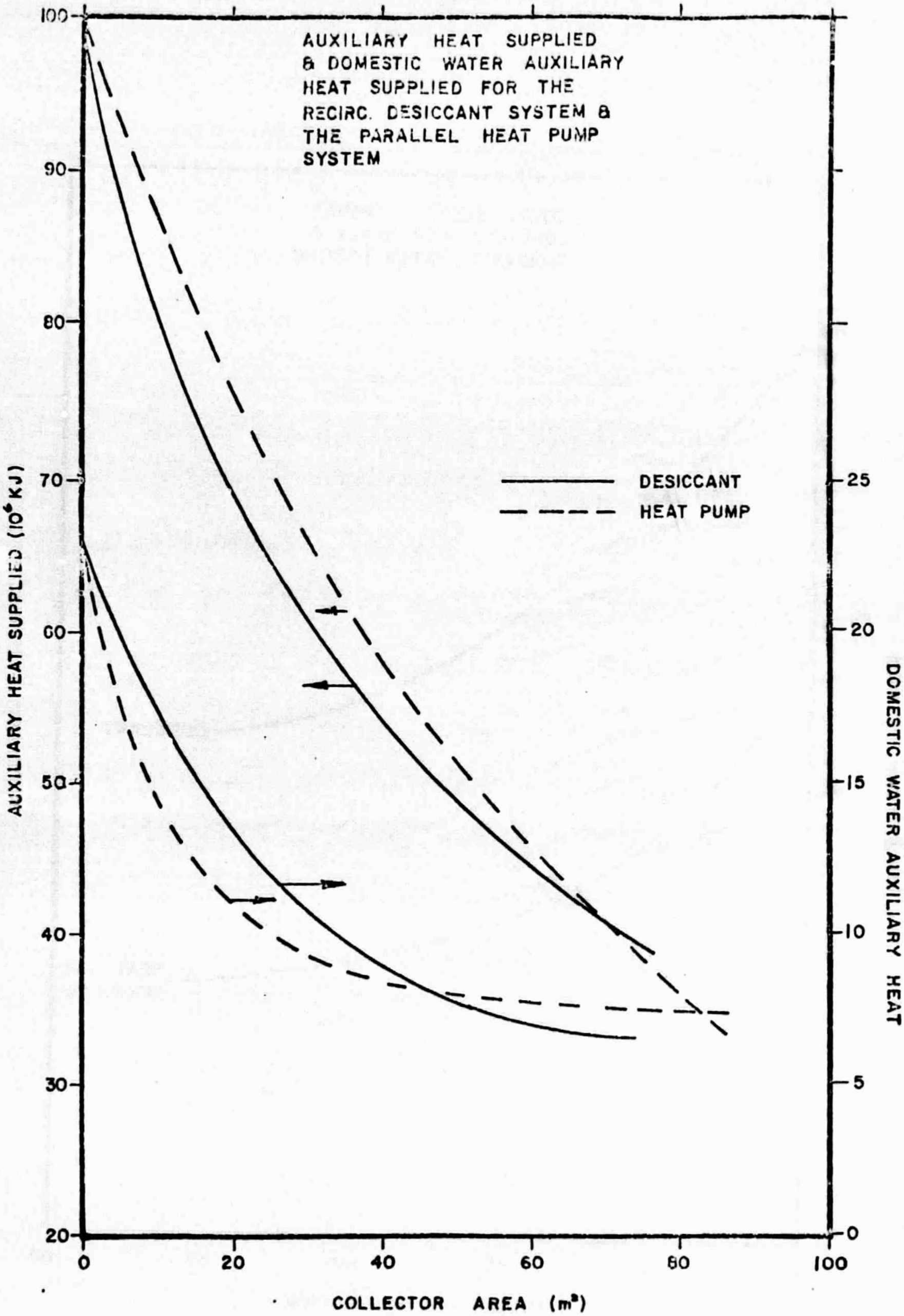



Figure 19 - Auxiliary heat supplied for space and domestic water heating versus collector area for the desiccant and parallel heat pump systems.

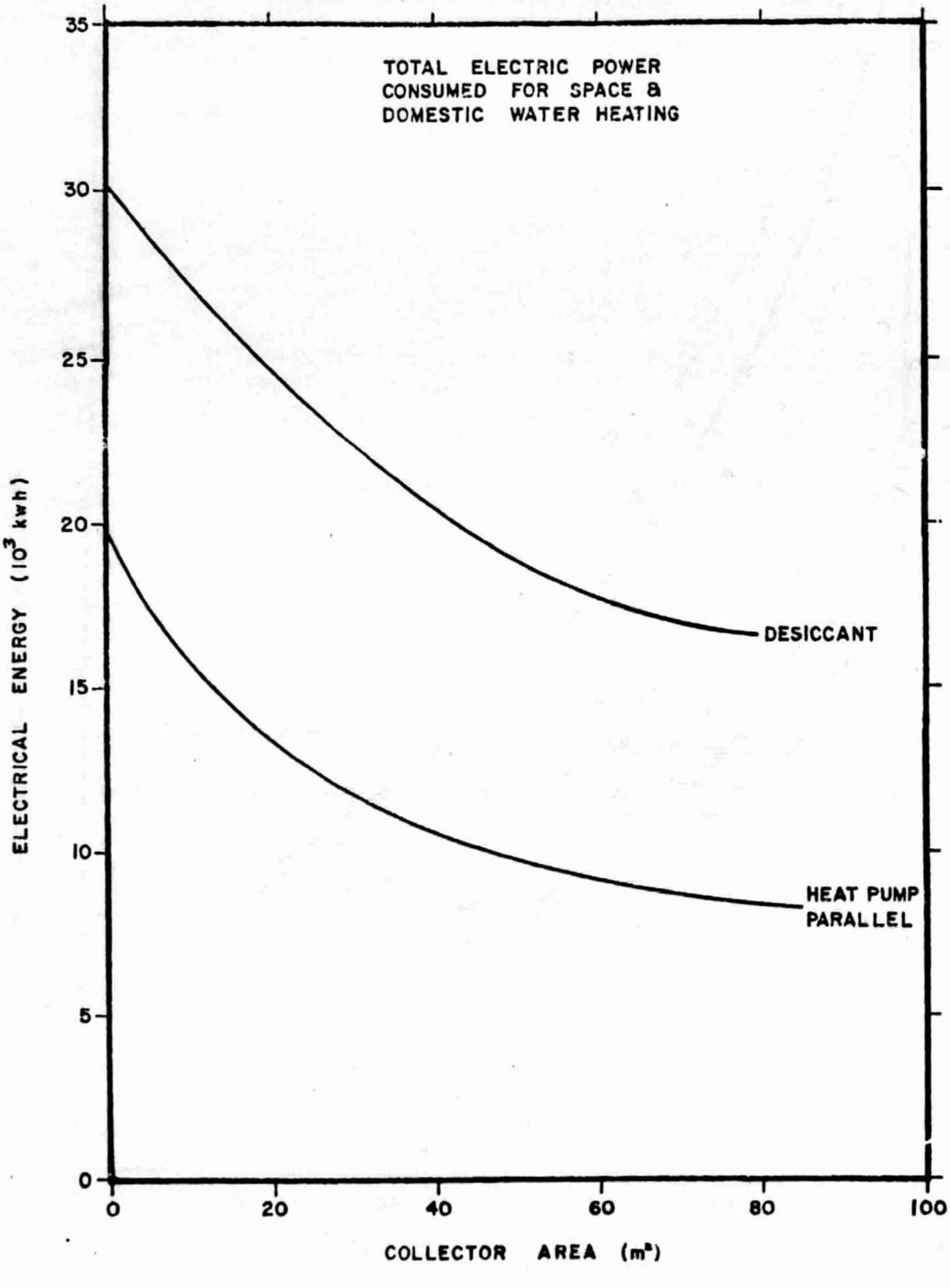



Figure 20 - Electrical energy used for space and domestic water heating versus collector area for the desiccant and heat pump system.

Figure 21 is a plot of the desiccant system percent of the total annual cooling load carried by solar and the recirculation cycle COP* for sensible heat exchanger effectivities of 85 and 90%. As is evident from the plot a system designed to carry 60-70% of total annual space and domestic water heating load by solar carries about 90% of the cooling load. Also evident is the benefit of the more effective (but more costly) sensible heat exchanger as both the percent solar and the cycle COP increase with the increased effectivity.

The total electrical energy consumed by the solar-desiccant cooling system is given in Figure 22. As is evident in the plot, the 90% effective sensible heat exchanger reduces the amount of energy needed for auxiliary and for fan power, the latter due to the more effective heat exchanger causing more cooling delivered per cfm and therefore shorter operating times.

Figure 23 shows a second cooling COP, the system cooling COP, versus collector area. The system cooling COP is defined as the cooling delivered divided by the total electrical energy input to the system, which includes the auxiliary heater and the fan power. As can be seen from the plot this system COP shows the expected rise as collector area increases, due to the auxiliary heater input decreasing.

In Figure 24 the total electric power consumed for cooling for the desiccant system is repeated from Figure 22. Also shown is the power consumed by the parallel heat pump system for cooling. As can be seen from the plot, the heat pump system uses less electricity for cooling than do the solar desiccant systems.

* Cycle COP is defined as the useful cooling delivered divided by the heat input (solar plus auxiliary) to the recirc. cycle.

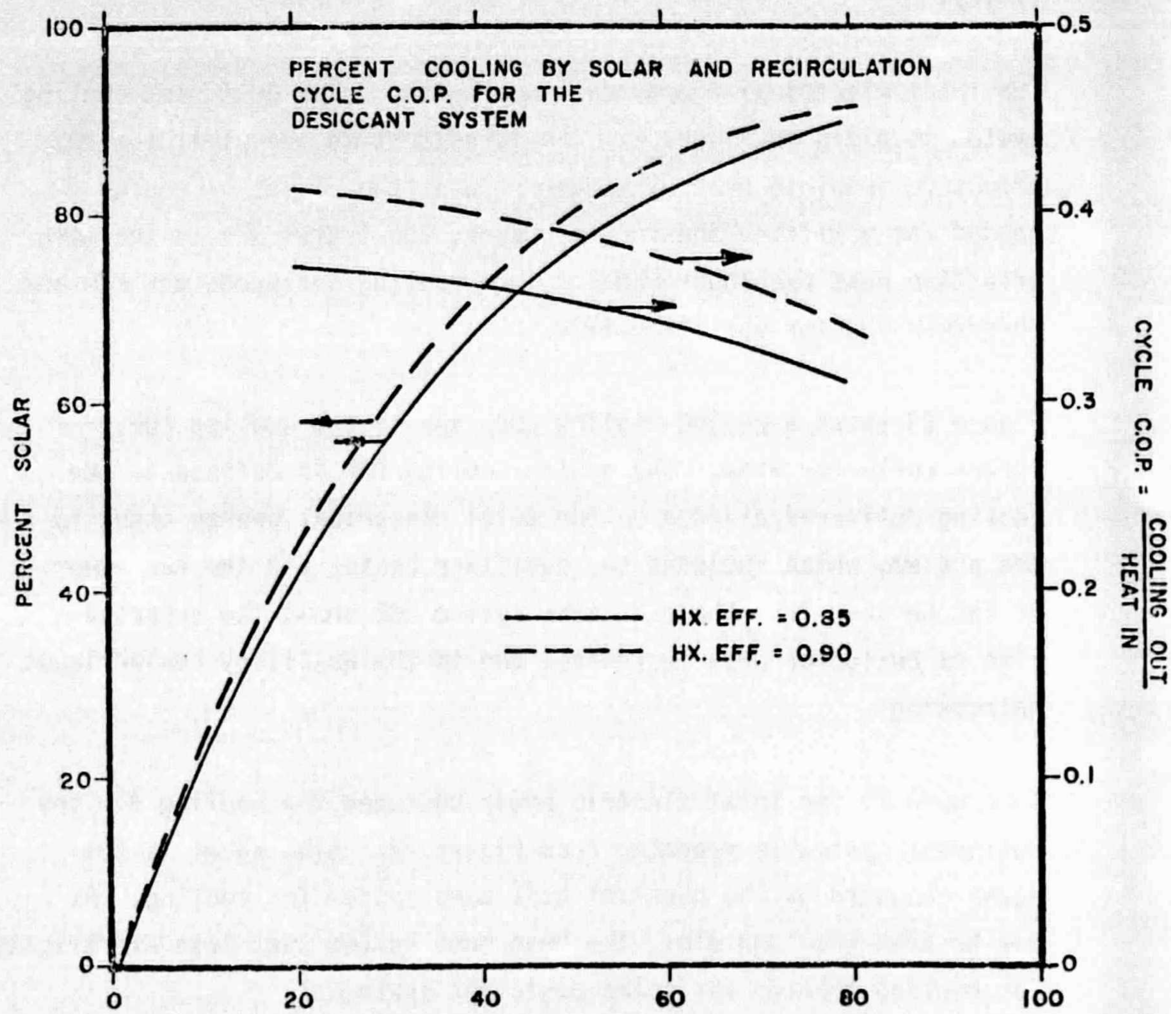



Figure 21 - Percent solar for space cooling and cycle COP versus collector area for the desiccant system at heat exchanger effectivities of 0.85 and 0.90.

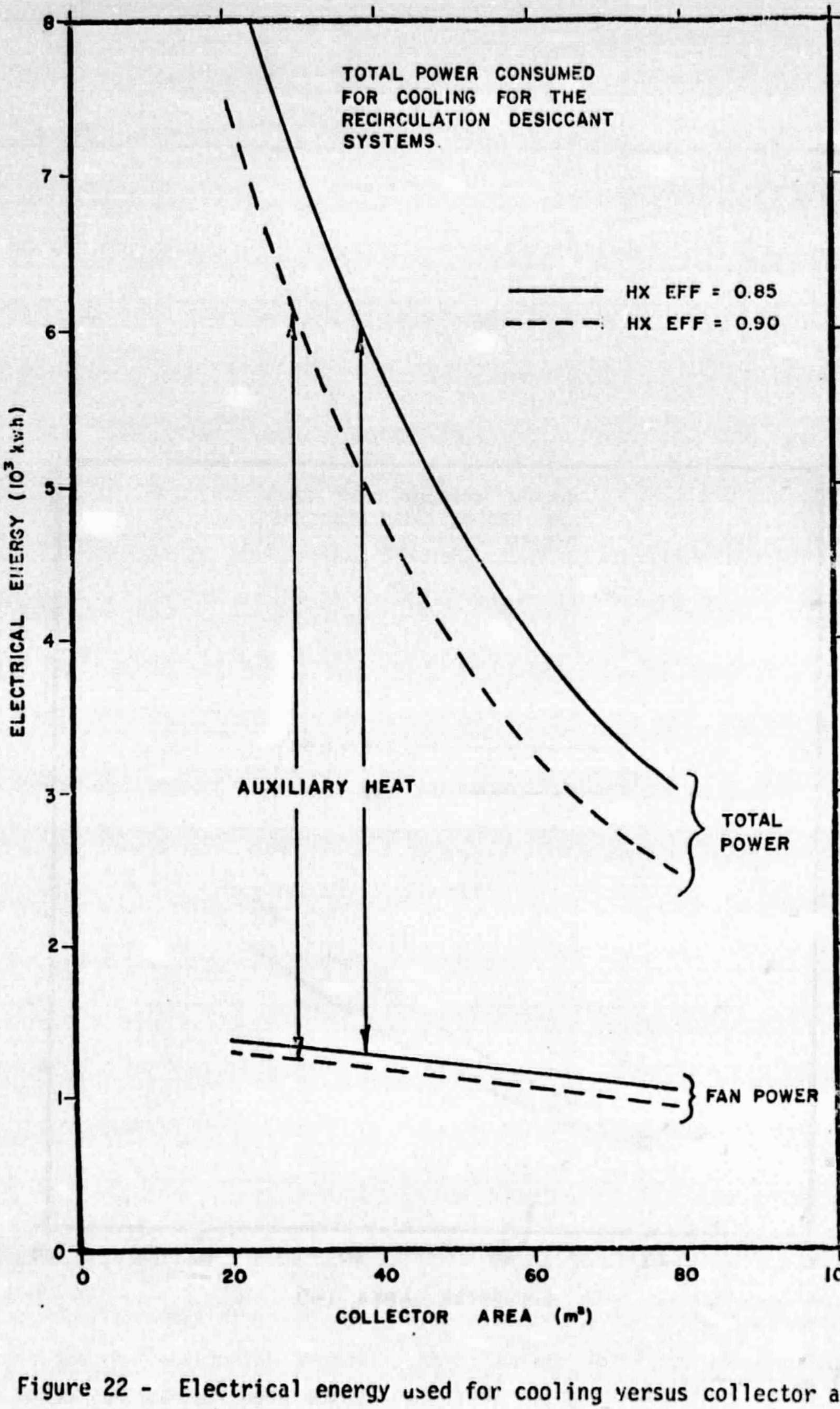



Figure 22 - Electrical energy used for cooling versus collector area for the desiccant system at heat exchanger effectivities of 0.85 and 0.90.

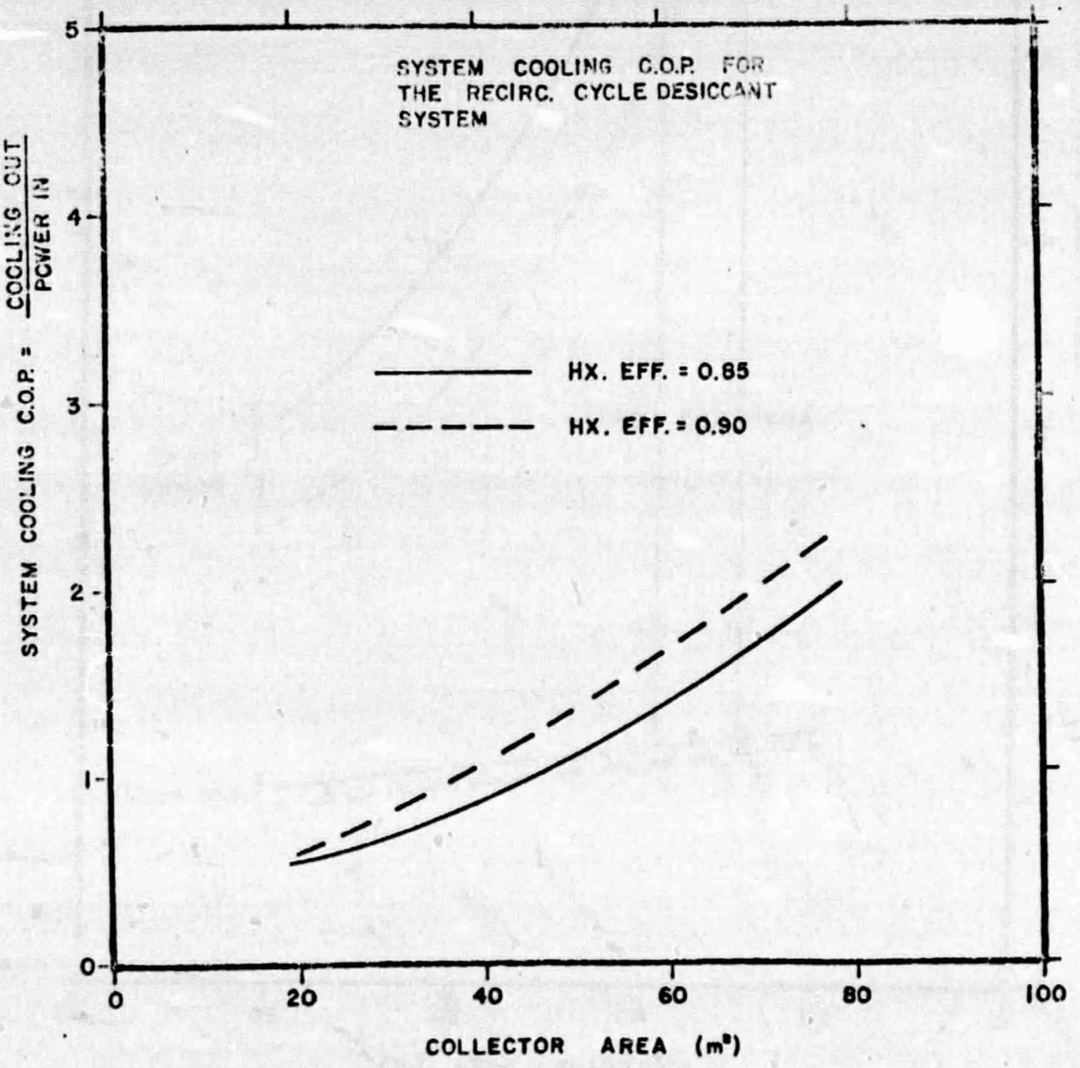



Figure 23 - System cooling COP versus collector area for the desiccant system at heat exchanger effectivities of 0.85 and 0.90.

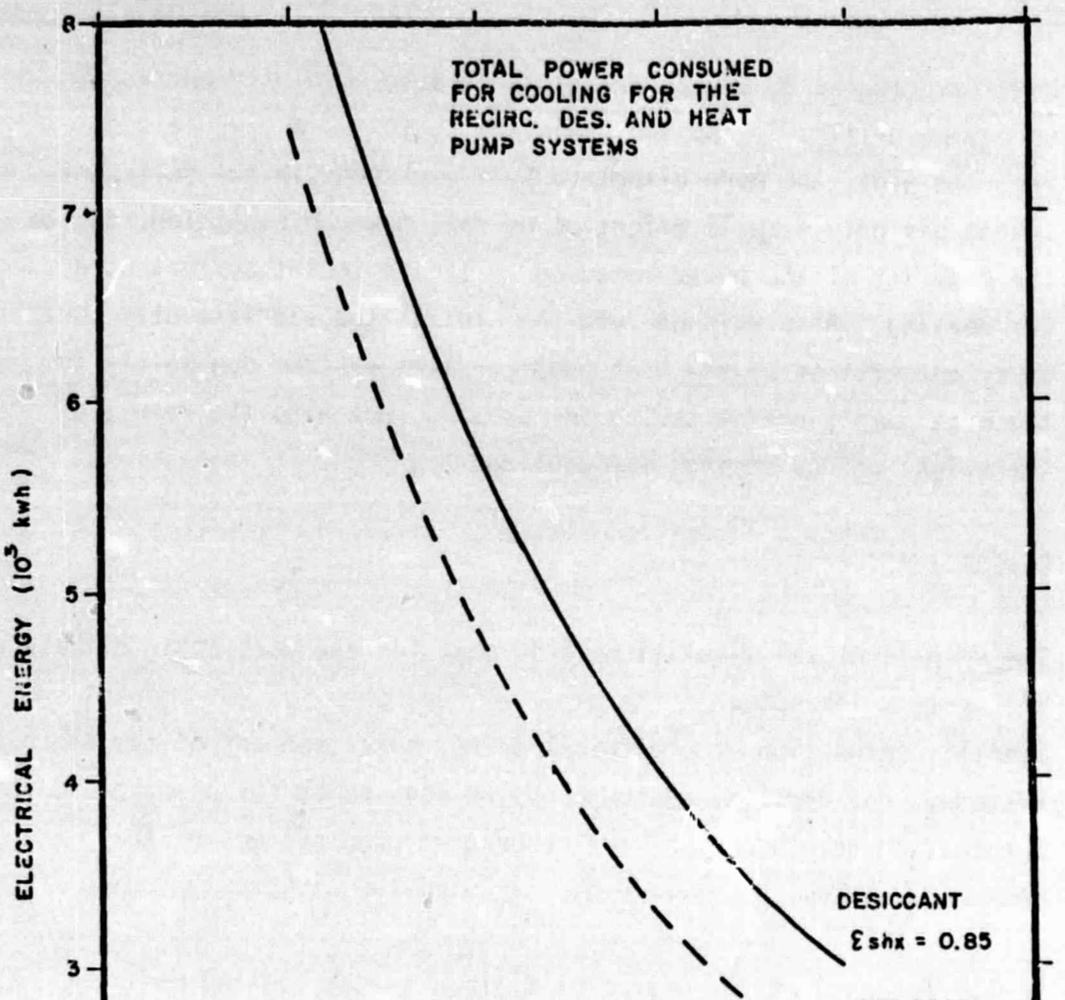



Figure 24 - Electrical energy used for cooling for the desiccant and parallel heat pump systems.

Finally, Figures 25 and 26 are plots of several of the pertinent energy quantities in the two desiccant systems. As can be seen from the plot, the more effective heat exchanger in the desiccant system has only a small effect on overall power consumption, due to the majority of the power consumed in the desiccant systems used for heating. Also evident from the plot is the significantly lower power consumption by the heat pump parallel system, due mainly to the heat pump's energy saving for heating, but also the result of the slight energy savings for cooling.

## III. Discussion

The results of the simulations show that for the particular climate and systems investigated the parallel heat pump systems reduces the overall consumption of electrical energy more than any of the other systems. The parallel system consumes less power for heating and for cooling than do any of the other heat pump systems or the desiccant system.

The performance of the desiccant systems in the cooling mode is somewhat disappointing. Due to the low recirculation cycle COP the desiccant systems when switched to auxiliary cooling consumed 7 to 10 times as much power per unit of cooling delivered as do the heat pump systems.

The possible methods of improving the overall performance of the desiccant systems lie mainly in the improvement of the cycle COP. The cycle COP can be improved several ways. One area of potential improvement lies in the control of the desiccant cycle. The control system employed is one originally developed by Nelson at the University of Wisconsin for the control of the ventilation desiccant cycle in a Miami simulation. This control is quite simple and for the Miami simulations gave cycle COP's for the vent cycle of 0.7 to 0.8, for a sensible heat exchanger effectivity of 90%. Development of a better control system of the New York situation is quite an intricate task which could take several months of analytical and experimental work.

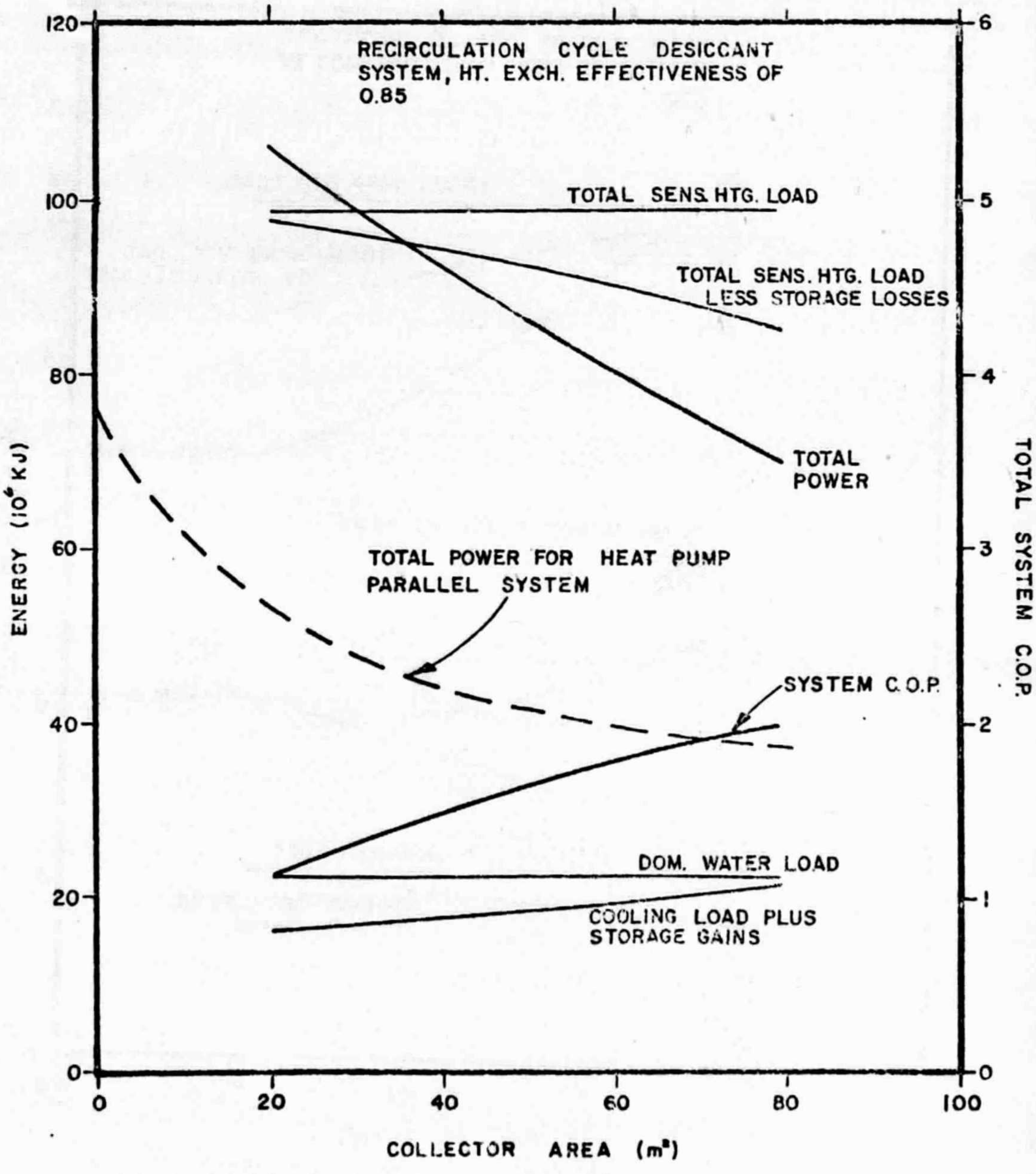



Figure 25 - Pertinent energy quantities and total system COP versus collector area for the 0.85 HX effectivity desiccant system.

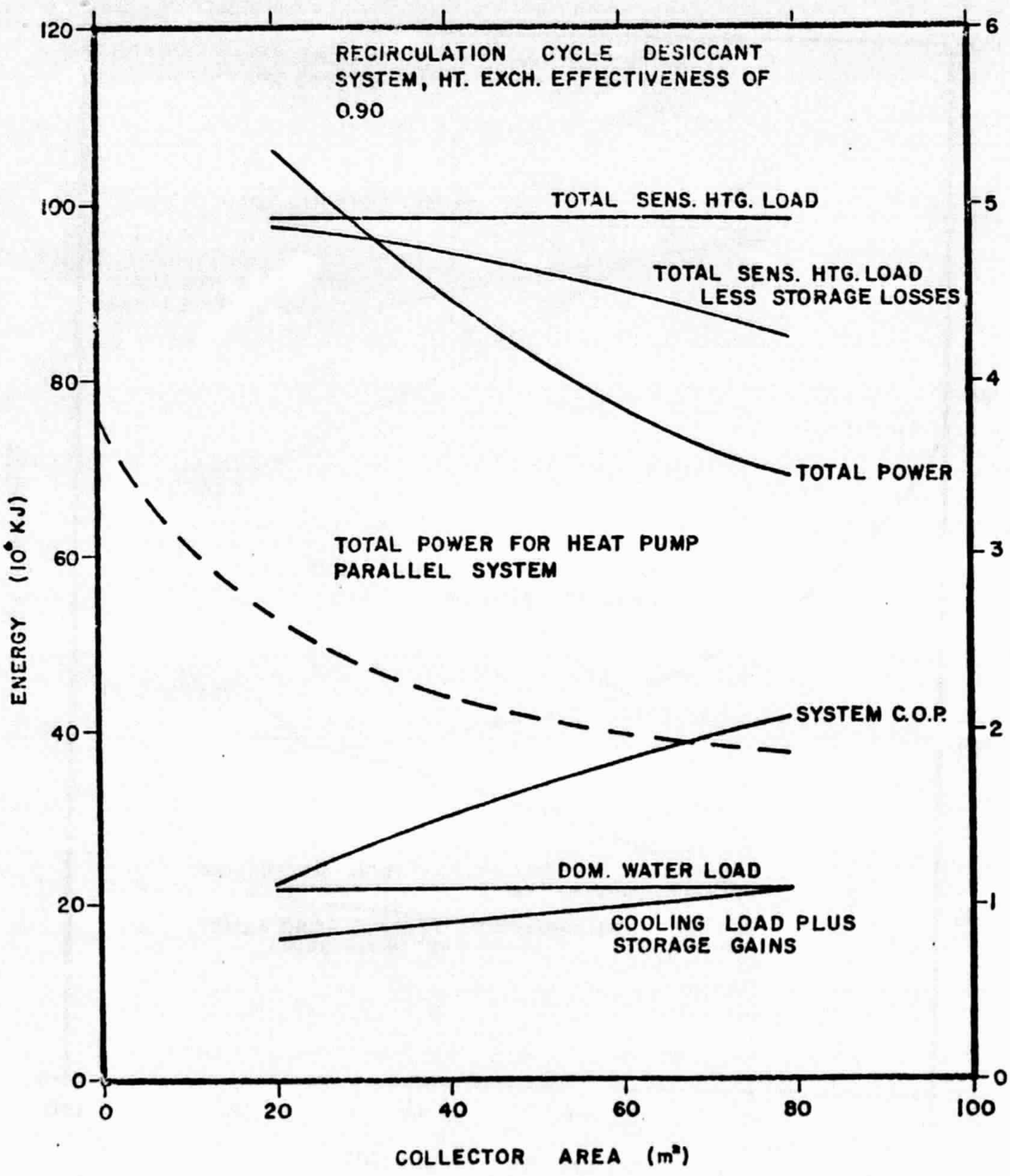



Figure 26 - Pertinent energy quantities and total system COP versus collector area for the 0.90 HX effectivity desiccant system.

A second area of improvement would be improvement of the individual component performances in the cycle. Of primary importance here would be improving the sensible heat exchanger performance. The difference in performance between the 85% effective and the 90% effective heat exchanger desiccant systems indicate that this small change in this component's performance results in a relatively large change in system performance. Another important improvement would be better desiccant wheel performance. Improvement in this area would probably not lead to as large of a change in system performance, however.

It is quite apparent that the heat pump system will save significant amounts of energy for heating over the resistance backup system used in the desiccant systems. It also appears upon final examination of all the available simulation results that there is no assurance that the proposed solar desiccant system will save fossil fuel as a cooling system over the parallel heat pump system. More development work on the desiccant systems (both analytical and experimental) is needed before the ability can be determined of improved designs of these systems to save fossil fuel.

APPENDIX

# SIMULATION OF FOUR OPEN CYCLE DESICCANT COOLING SYSTEMS

Report Submitted To

SOLARON CORPORATION

by

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Patrick J. Hughes

William A. Beckman

Solar Energy Laboratory
University of Wisconsin
Madison, WI

June 20, 1977

Notes: (By Solaron)

1. Computer Simulation for this study is based on hourly weather data for New York City, New York from 1 July through 22 Sept. 1957.

   Total year data is available from 1 July 1957 through 30 June 1958.

2. Psychrometric cycles have been added to correspond with the various system schematics (Figures 3, 5, 7 & 9).

3. Figure 21 added to show integration of solar heating and cooling systems.

4. Figures and numbers on the figures themselves have been added/revised (Figures 1 through 21), but reference numbers have not been changed in the text written by the University of Wisconsin.

5. A descriptive narrative has been added for figures 1 through 21, and Tables 1 & 2.

6. The COP line of Figure 16 is now plotted correctly. As presented at the TR/QRM meeting of 27 June 1957 the COP line was erroneously plotted against the wrong scale. The tabular data is correct.

7. Coefficient of performance (COP) as shown and used in these figures is based on the total energy input to the desiccant cycle, including both conventional and collected solar heat and is equal to total cycle cooling output divided by total heat input to the cycle. This parameter is useful in a comparative evaluation of the various cycles.

   For evaluating systems relative to conventional energy usage, the COP should be evaluated as total system output divided by conventional energy input only. These values of COP will be generated in future analyses.

## DESCRIPTION OF SYSTEMS

The general arrangement of the open cycle, desiccant cooling system using solar energy simulated in these studies is shown schematically in Fig. 1. The solar energy collection system is a Solaron system and the space heating load is that of the HUD AAD cluster for New York City, New York.

The solar desiccant system is built up of available components in four configurations. These are shown schematically in Figures 2-5. The control strategy is the same as that used by J. R. Nelson in his thesis for the Miami simulations.

Preliminary calculations were done with f-chart to determine reasonable sizes for the collector. It was found that 50 $m^2$ of collector allowed 40% of the space heating load to be met, 100 $m^2$ met 60%, and 135 $m^2$ met 70%.

## SYSTEM PARAMETERS

The major parameter values used for all system simulations are given below.

Collector

$F' = 0.781$

$U_L = 0.85$ Btu/hr ft$^2$F

$\alpha = 0.95$

$\tau = 0.815$

$\overline{\tau\alpha} = 0.753$

Rock Bed

$L = 6$ ft

$A = 0.125\ A_c$

Rock Bed, cont'd.

$U = 0.0832$ Btu/hr $ft^2F$

$k = 0.022$ Btu/hr ft F

$C_r = 0.22$ Btu/lb F

$\rho_r = 95$ $lb/ft^3$

Domestic Hot Water System

Heat Exchangers: UA = 440 Btu/hr F

Tank: V = 80 gallons

$U = 0.1$ Btu/hr $ft^2F$

Water Flow Rate: Scheduled over period 6 AM to 7 PM

Total flow is 667 lb/day

Desiccant Systems

Dehumidifier: Area/Vol = 1028 $m^2/m^3$

Flow Length = 0.0381 m

Total Frontal Area = 1.56 $m^2$

Void Fraction = 0.4

Rotational Speed = $1.023 \times 10^{-3}$ rev/sec

Conductance = 121 $w/m^2K$

Friction Factor = 0.9

Desiccant Mass/Air Mass = 1607

Sensible Heat Regenerator:

Effectiveness = 0.90

Evaporative Coolers

Maximum Effectiveness = 0.90

Indirect Evaporative Cooler Effectiveness = 0.95

House Load

New York City weather for July 1 to September 22, 1955

House UA = 1700 kJ/hrC

Thermostat Setting = 23.8C

Humidistat Setting = 0.009 Kg/Kg

Sensible Load = $2.0 \times 10^6$ W-hr

Latent Load = $1.2 \times 10^6$ W-hr

(The loads were found to be constant for all systems simulated.)

RESULTS

Three sets of simulations were performed as follows:

Set A: Ventilation mode, recirculation mode 1 and recirculation mode 3 for a constant air flow rate per unit collector area of 2 cfm/$ft^2$ and for variable collector area.

Set B: All four modes for a constant mass flow rate of 1540 cfm and variable collector area.

Set C: All four modes for a constant collector area and variable flow rate.

The results for Set A are given in Table 1 and presented graphically in Figures 6, 7 and 8. Note that there are no entries for the water consumption. A programming error occurred, and the previous values reported are wrong. The coefficient of performance is the total load met divided by the sum of the solar and auxiliary energy supplied.

The results for Sets B and C are given in Table 2 and shown in Figures 9, 10, 11, and 12. The results are grouped by mode to show the influence of area and flow rate on each mode. For the recirculation mode, option 3, the water flow rate does not include that used in the indirect evaporative cooler. The amount of water used was not calculated as the evaporative cooler was assumed to have a constant 95% effectiveness.

## CONCLUSIONS

The following are conclusions that can be drawn from this study.

1. The results from Set A show that the flow rate through the machine and load is important. Either too large or too small a flow rate requires additional auxiliary energy. For large collector areas, it is better to reduce collector performance by lowering flow rate than to maintain a design value of flow per unit collector area.

2. The results from Set B and C show that the solar system can provide essentially all of the energy required. The collector area required for 90% solar depends on mode as follows:

| | $A_c(m^2)$ |
|---|---|
| Ventilation | 60 |
| Recirculation, Option 1 | 55 |
| Recirculation, Option 2 | 40 |
| Recirculation, Option 3 | 40 |

The implication is that for these collector areas and larger, the systems will probably operate satisfactorily with no auxiliary and acceptable diurnal changes in room temperature.

3. The coefficients of performance are somewhat lower than expected, and show a decrease with increased area. This is due, in part, to the available solar temperature being much greater than needed much of the time. Thus, solar energy is not effectively used. Improved control could alleviate this situation.

4. The water useage for all modes is comparable. The average useage is about 50 kg/day, or 10-15 gal/day.

5. The hours of operation decrease with increased area. The machine is operating six hours per day on the average. However, some days it is on more than 12 hours, and others on as little as one hour, depending on the weather for that day.

6. The results of Set C show that, generally, the lowest flow rate yields the best performance. The ventilation mode is the most sensitive to flow rate. Option 3 shows a slight decrease in auxiliary with increased flow.

7. The pressure drops through the dehumidifier and regenerators are about 0.1 and 0.07 inches water, respectively, per side. The overall pressure drop through a desiccant system with one dehumidifier and one regenerator at 1540 cfm is then about 0.34 inches water. For 400 hours of operation the fan power would be about 24,000 w-hr, or about 0.8% of the load.

8. The control strategy needs considerable further research before an optimum design is obtained. The strategy employed is that used in Nelson's thesis for Miami, and is obviously not satisfactory for a climate like New York. An actual system could undoubtably be made to operate better using an improved control strategy.

## NARRATIVE DESCRIPTION OF FIGURES

Fig. 1 This figure is a schematic of a Solaron heating system integrated with any of the proposed desiccant air conditioning systems. The basic modes of operation of the system allow the desiccant system to be regenerated directly from the collector or from the stored heat in the pebble bed. The system can also store heat from the collector in the pebble bed, for periods when there is solar collector heat available and there is no simultaneous need for cooling.

Fig's. 2 through 9 This group of figures shows the variety of solar assisted desiccant cooling system modes presently under study and development by Solaron Corporation. Each mode is described by a system flow schematic (Fig's. 2, 4, 6 & 8) followed by a corresponding psychrometric cycle diagram (Fig's. 3, 5, 7 & 9).

Fig's. 2 & 3 Vent Cycle - This proposed cycle is called a vent cycle, due to its use of 100% fresh air. Referring to the system flow diagram and psychrometric chart, ambient air at state 1 is drawn through a desiccant wheel, where it is dried and heated to state 2. The air then is cooled by a sensible air-to-air heat exchanger to state 3 followed by evaporative cooling to state 4, where it enters the room. Return air from the room (state 5) is first evaporatively cooled to 6 and then is heated by the sensible heat exchange process to state 7. At this point it is further heated by the solar system (either by the collector or storage, Fig. 1) and an auxiliary heater if needed, to state 8. This hot air then regenerates the desiccant wheel and is discharged to the atmosphere at state 9.

Fig's. 4 & 5 Recirculation Cycle, Option 1 - This is one of three proposed recirculation cycles. Recirculation in this case refers to the room air stream flowing in a closed loop, while a second open loop air stream is used for regeneration. Referring to the room air loop first, return air at 6 is drawn through a desiccant wheel where it is dried and heated to 7; the hot dry air at state 7 is then sensibly cooled to state 8 and evaporatively cooled to state 9 where it re-enters the rooms. The regeneration cycle consists of ambient air at state 1 which is evaporatively cooled (state 2) and then sensibly heated (state 3). Air at state point 3 is then heated by the solar system and/or auxiliary (if required) to state 4; this air then is used to regenerate the desiccant wheel and is discharged to atmosphere at state 5.

Fig's. 6 & 7 Recirculation Cycle, option 2 - In this cycle a slight change is made to the system in option 1 in an attempt to increase the overall COP. Return air at state 5 is dried to state 6 and sensibly cooled as before, however at state 7 the air is sensibly cooled further by an indirect evaporative cooler, which allows state 8 to closely approach the outdoor wet bulb temperature. Air at state 8 is then evaporatively cooled to 9 where it enters the room. Ambient air used in the regenerative loop actually takes two paths. A relatively large flow is used in the indirect evaporative cooler to provide cooling to the room air stream, while an additional amount of air equal in flow to the room air rate is directed into the sensible heat exchanger. This air is heated to the same temperature at state 2 as was the air at state 3 of option 1, but since it has not been evaporatively cooled in this cycle it is at a lower humidity. This allows the air at state 3 of this cycle to be at a lower temperature and still achieve the required regeneration of the wheel.

Fig's. 8 & 9 Recirculation Cycle, option 3 - This cycle is exactly the same as option 1 on the room side, but is significantly different on the regeneration side. Specifically, ambient air enters at two different points in the regeneration process. One ambient air stream is evaporatively cooled from state 1 to 2 and then heated from 2 to 3 by the first sensible heat exchanger. Ambient air at state 1 also enters a second sensible heat exchanger and is heated to state 4; this air is then further heated by the solar system and/or auxiliary to state 5. This air then regenerates the wheel and leaves at state 6 where it enters the second sensible heat exchanger, is cooled and is discharged at state 7.

Tables 1 & 2 The results of the three sets of Simulations (Sets A, B & C) are presented in tabular form in these Tables. Table 1 gives the results of Set A, and Table 2 gives the results of Sets B & C.

Fig's. 10 through 20 present graphically the preliminary results of a computerized cycle simulation program at the University of Wisconsin. This particular study was run on a specific house in New York City, N.Y. using hourly weather data recorded in the period of 1 July through 22 September 1957. In this way comparison of various cycle parameters for the modes and system cycles is directly related to the same set of operating conditions and loads. The parameters analyzed in this study are COP, hours of system operation, water consumption, and auxiliary energy usage, all totalized over the entire period of the study, 1 July through 22 September.

Figs. 10, 11 & 12 present the above data (Set A) for a standard (normal) air flow rate of 2 SCFM/ft.$^2$ (21.52 SCFM/meter$^2$) of Collector area, and for variable collector area.

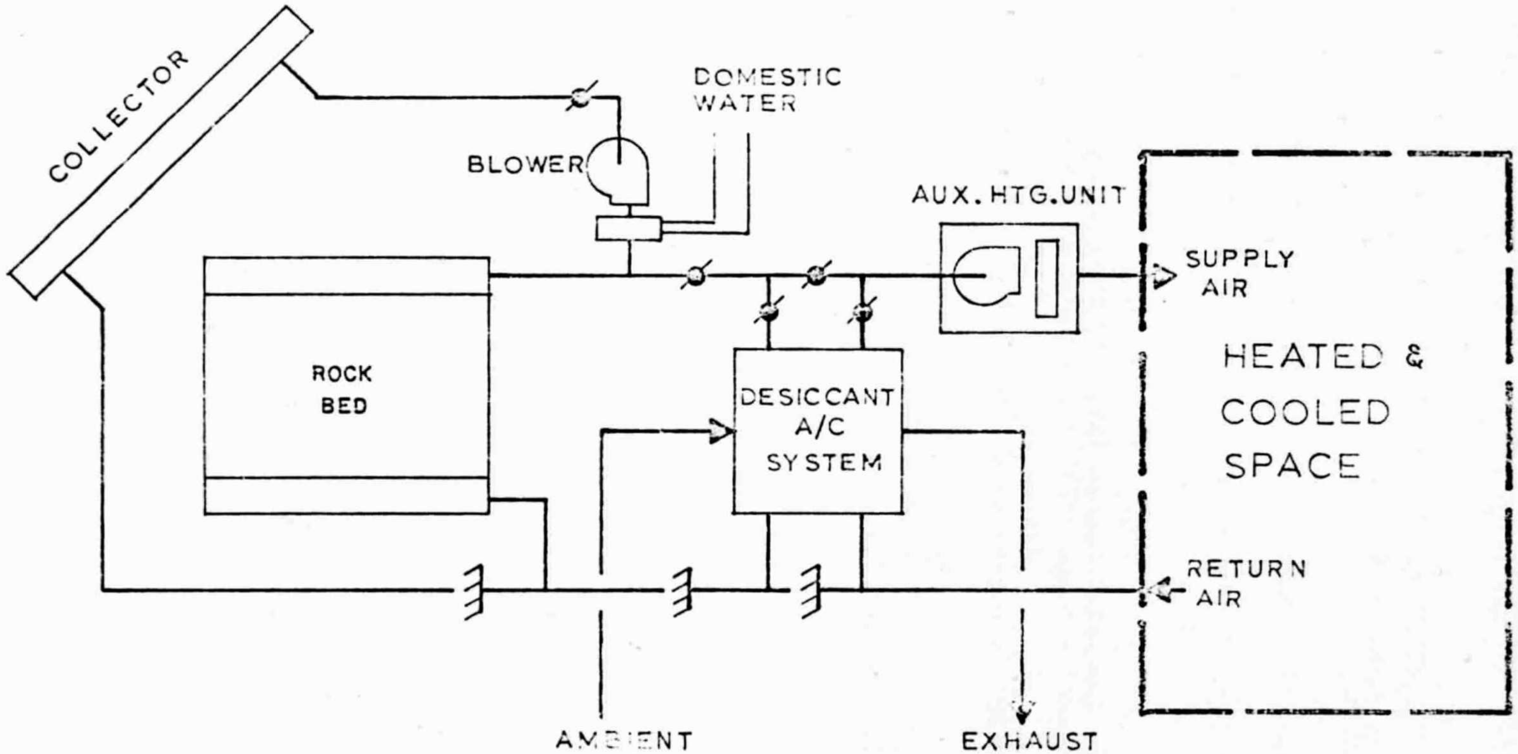
Fig's. 13, 15, 17 & 19 present the above data (Set B) for a constant air flow rate of 1540 SCFM and a variable collector area from 25 to 75 square meters.

Fig's. 14, 16, 18 & 20 present the same data (Set C) for a constant collector area of 55 square meters and a variable air flow rate from 1260 to 1680 SCFM.

Fig. 21 This is a schematic flow diagram for an integrated solar heating and cooling system with additional provision for domestic hot water. The cooling portion of this system is Solaron desiccant recirculation cycle, option 1, as shown in Fig's. 2 & 3.

COP - Coefficient of Performance (COP) as shown and used in these figures is based on the total energy input to the desiccant cycle, including both conventional and collected solar heat and is equal to total cycle cooling output divided by total heat input to the cycle. This parameter is useful in a comparative evaluation of the various cycles.

For evaluating systems relative to conventional energy usage, the COP should be evaluated as total system output divided by conventional energy input only. These values of COP will be generated in future analyses.



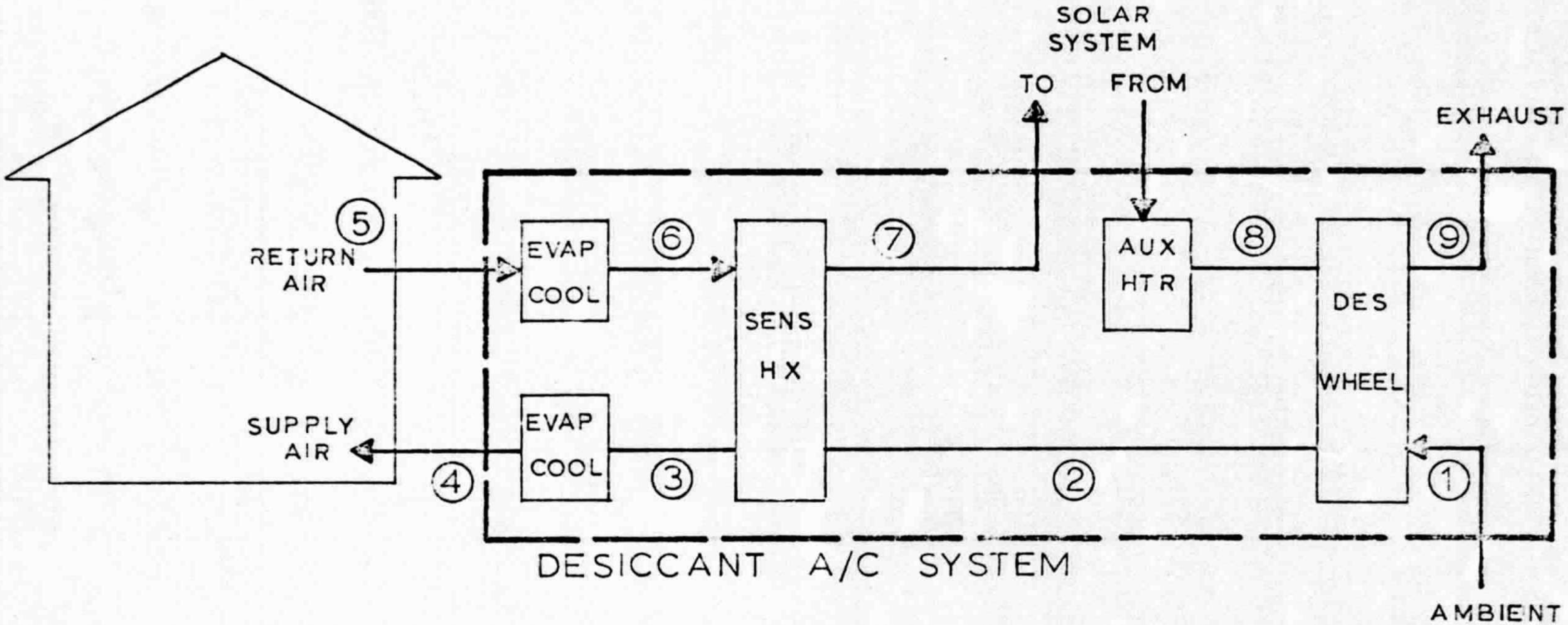

SOLARON System Integrated With Desiccant Air Conditioning System

MODES:

HEATING OPERATION
- HEAT FROM COLLECTOR
- STORE HEAT
- HEAT FROM STORAGE

COOLING OPERATION
- REGENERATE FROM COLLECTOR
- STORE HEAT
- REGENERATE FROM STORAGE

FIG. I




VENT CYCLE

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

FIG. 2

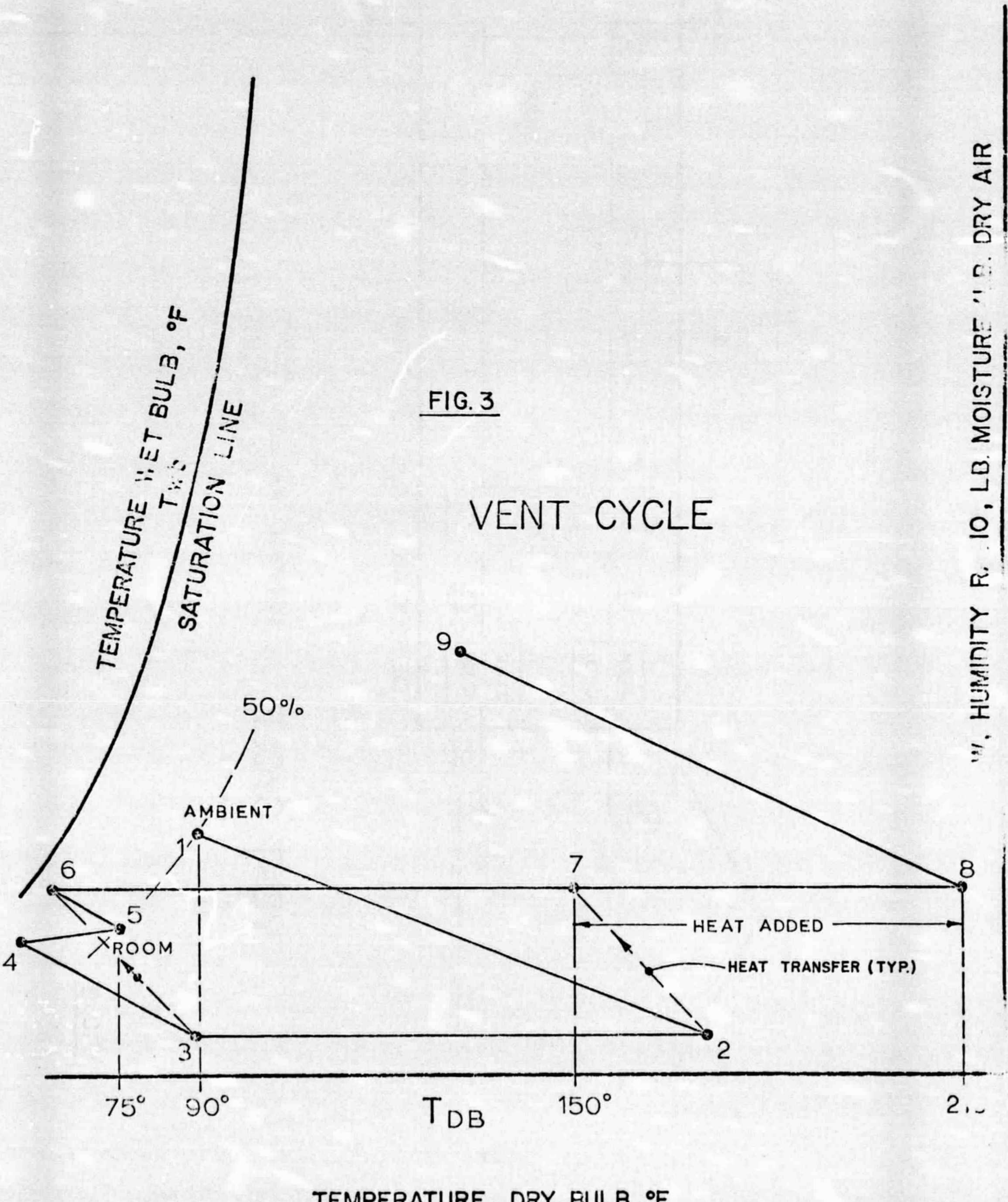
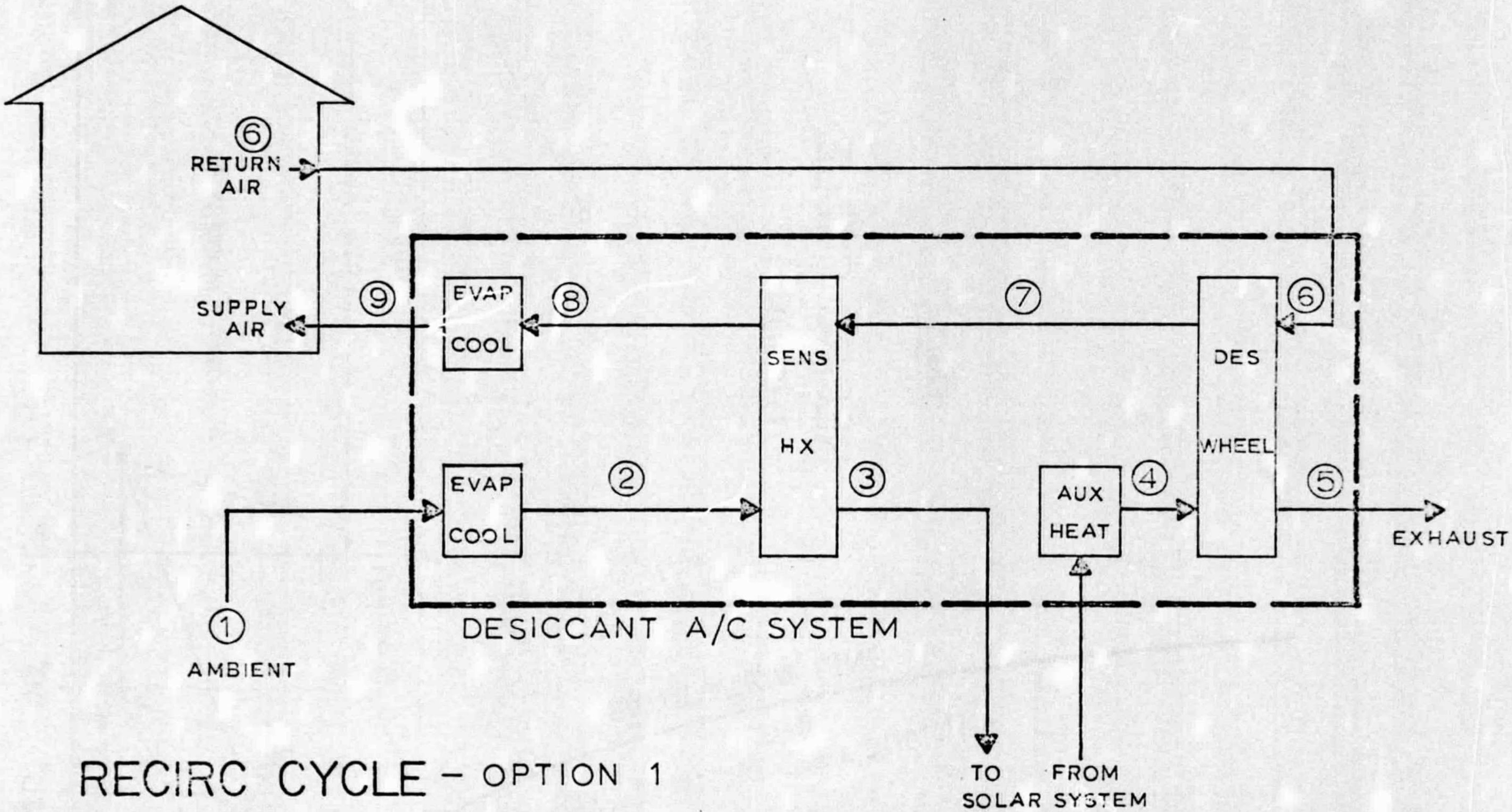



FIG. 3

VENT CYCLE

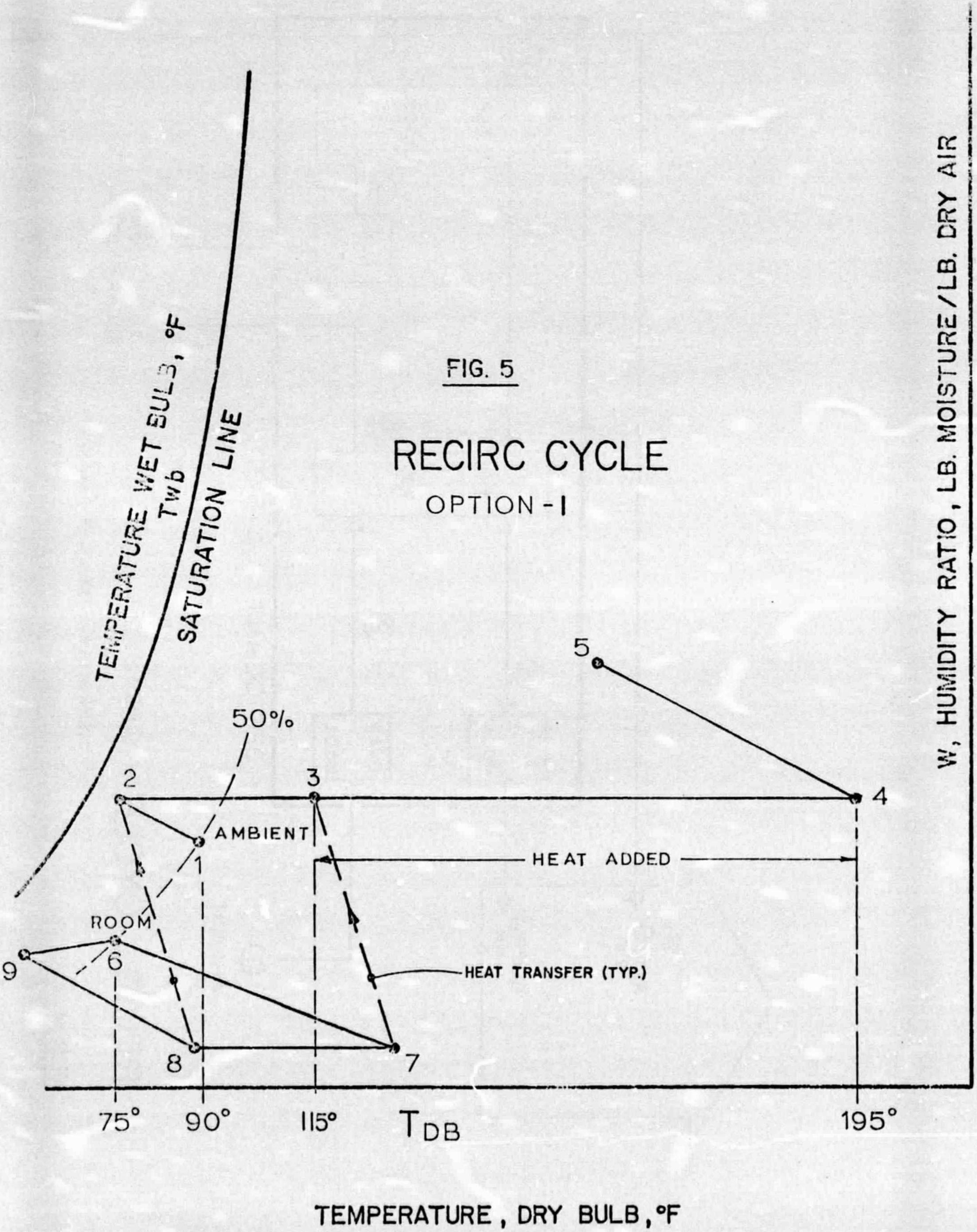


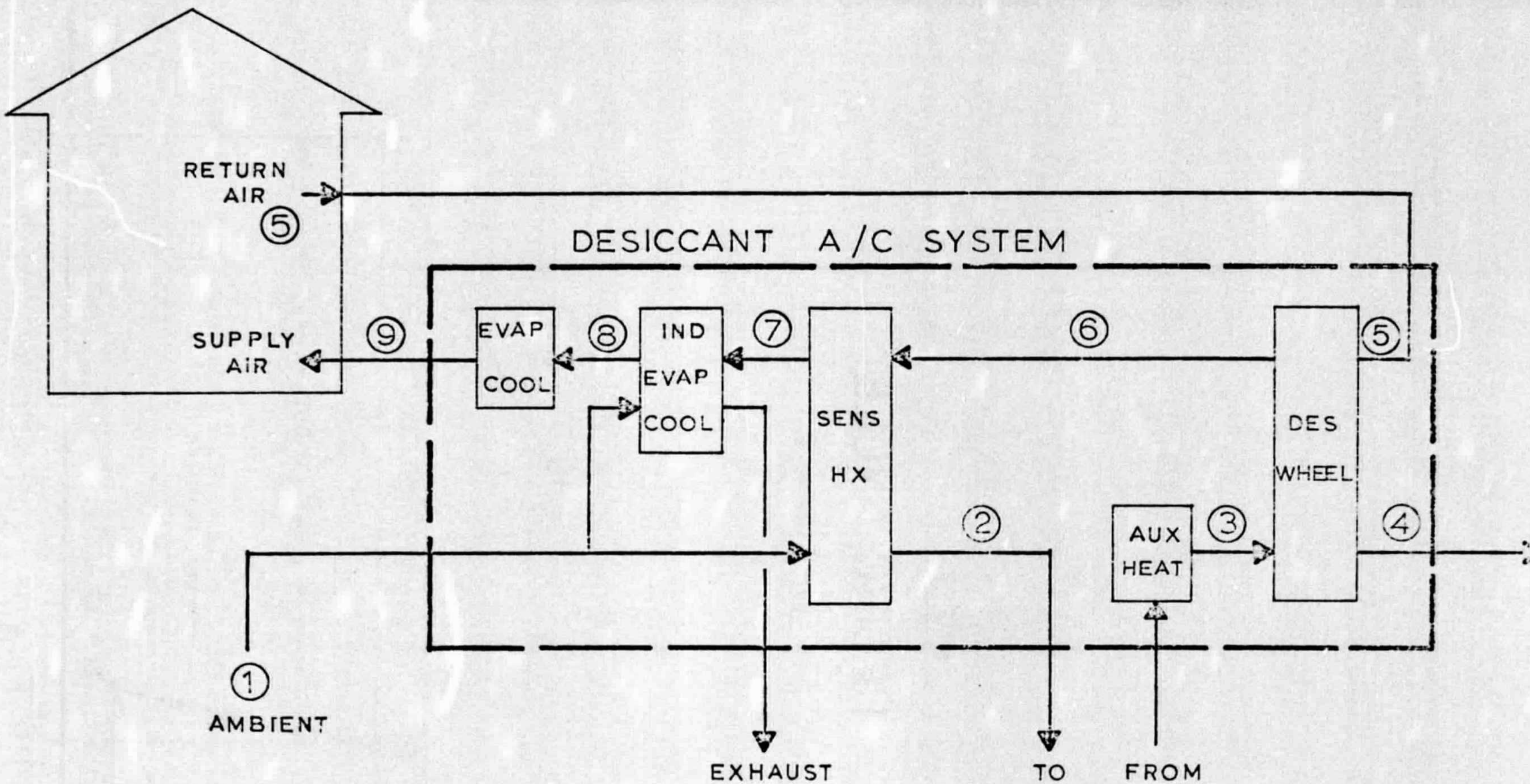

RECIRC CYCLE - OPTION 1

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

FIG. 4


TEMPERATURE WET BULB, °F
Twb
SATURATION LINE
FIG. 5
RECIRC CYCLE
OPTION - I
W, HUMIDITY RATIO, LB. MOISTURE/LB. DRY AIR
50%
5
2
3
4
AMBIENT
1
HEAT ADDED
ROOM
9
6
HEAT TRANSFER (TYP.)
8
7
75°
90°
115°
TDB
195°
TEMPERATURE, DRY BULB, °F




RECIRC CYCLE

OPTION 2

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

INDIRECT EVAP COOLER $\eta = \dfrac{T_{DB7} - T_{DB8}}{T_{DB7} - T_{WB1}} = 0.63$

$= 0.86$

$= 0.95$

FIG. 6

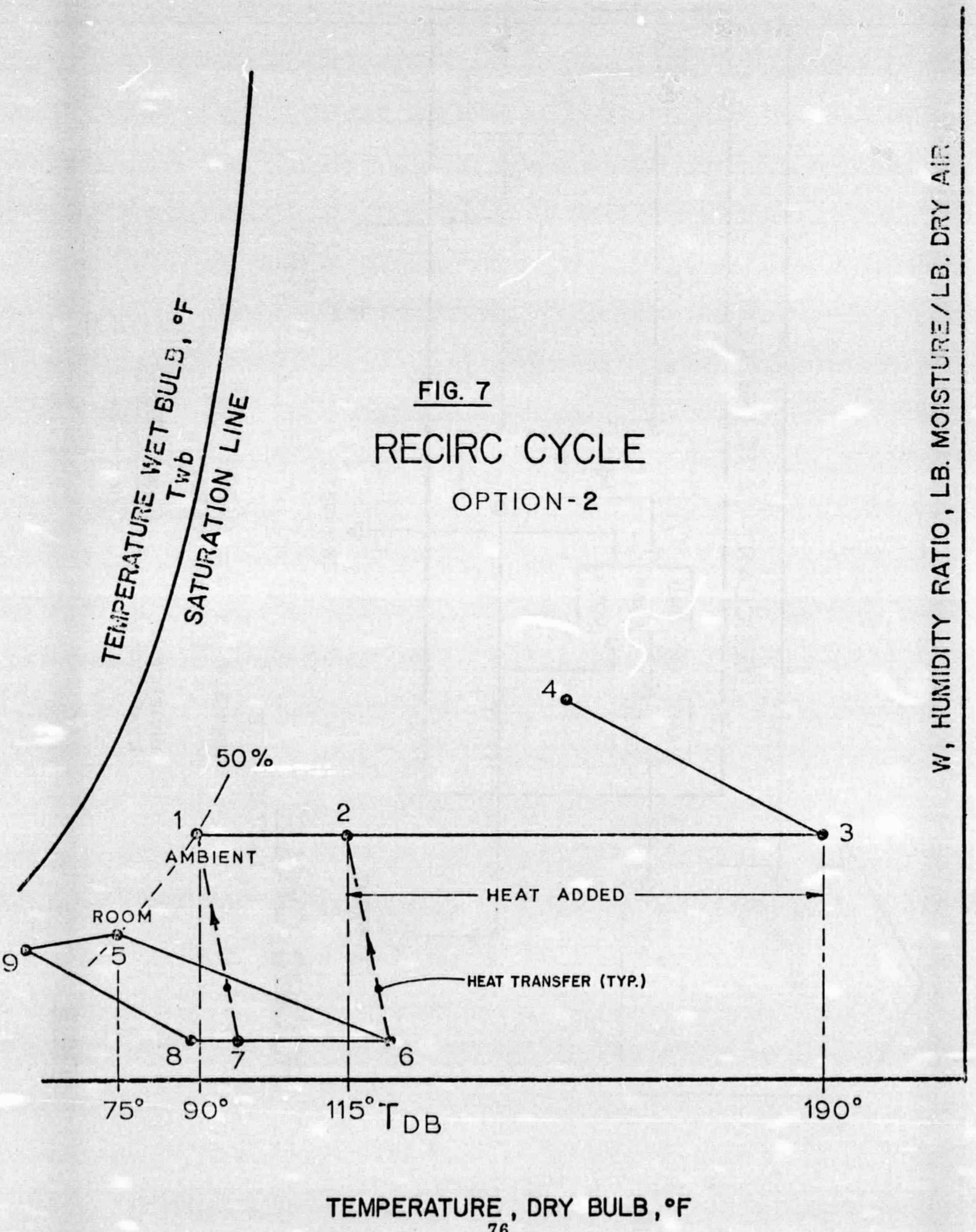
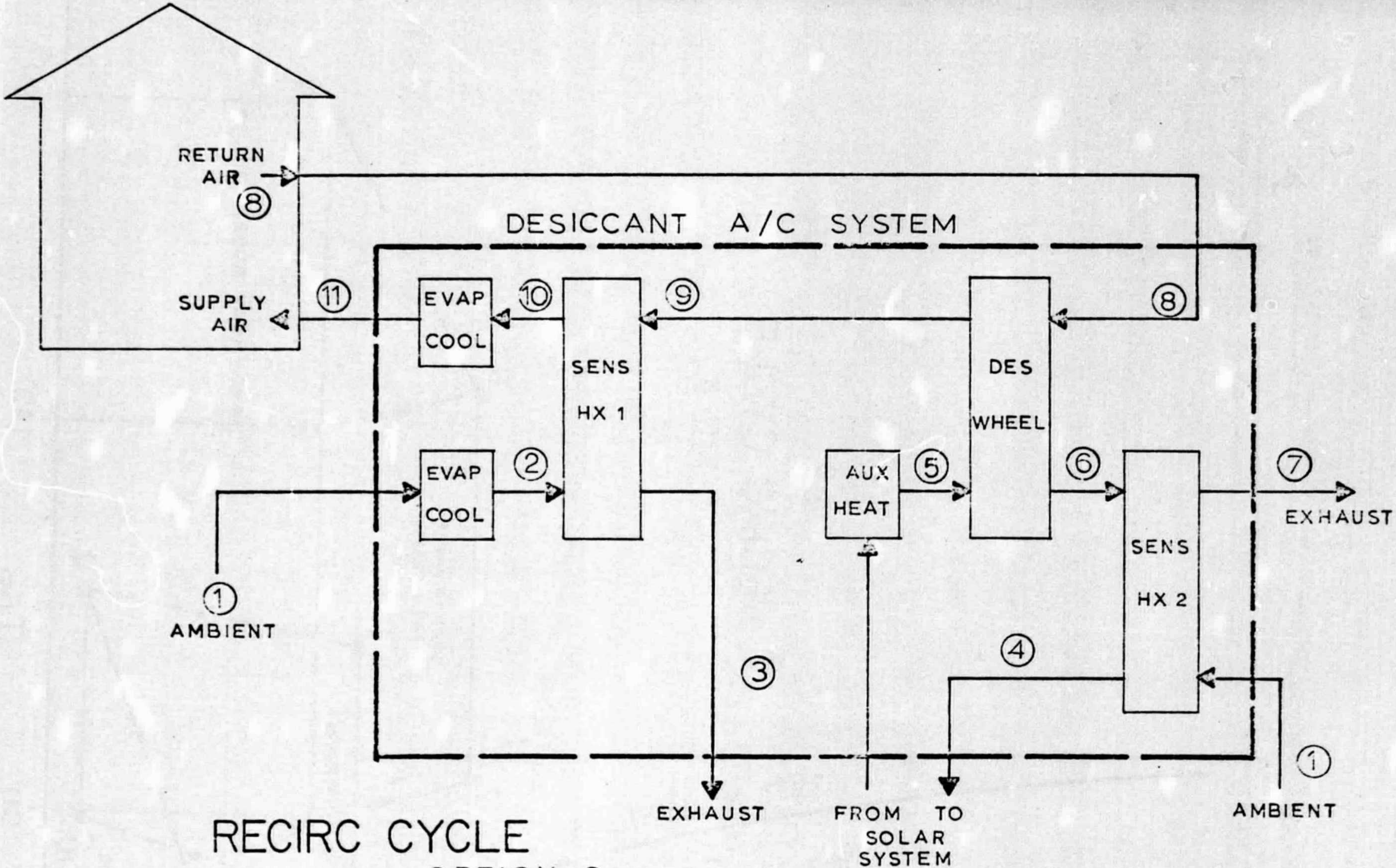

TEMPERATURE WET BULB, °F
Twb
SATURATION LINE
RECIRC CYCLE
OPTION-2
W, HUMIDITY RATIO, LB. MOISTURE/LB. DRY AIR
50%
1
AMBIENT
2
3
4
HEAT ADDED
ROOM
5
9
HEAT TRANSFER (TYP.)
8
7
6
75°
90°
115°
$T_{DB}$
190°
TEMPERATURE, DRY BULB, °F

FIG. 7




RECIRC CYCLE
OPTION 3

SENS HX 1 $\varepsilon = 0.70$
$\varepsilon = 0.85$
HX 2 $\varepsilon = 0.70$
$\varepsilon = 0.85$

FIG. 8

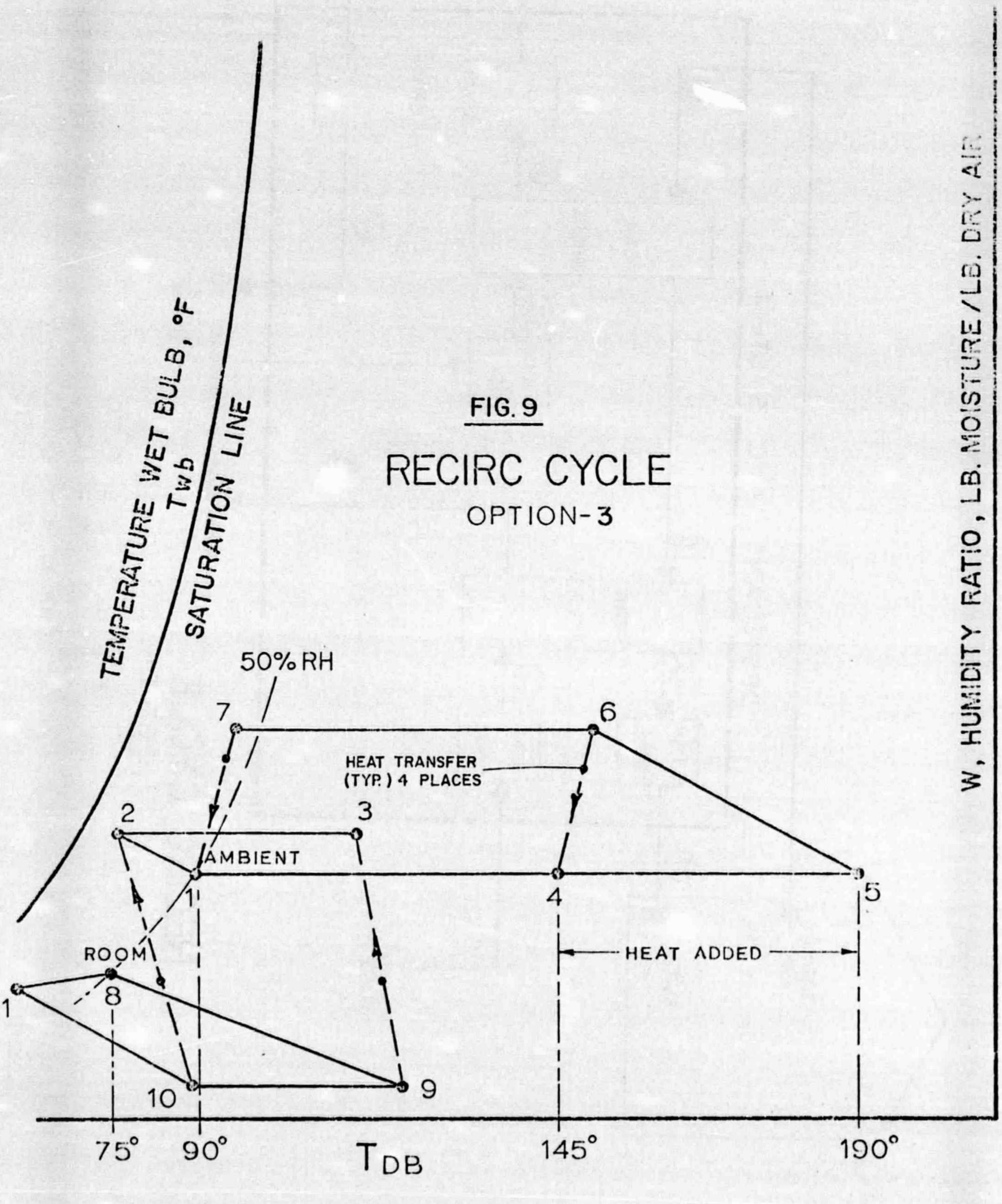

TEMPERATURE WET BULB, °F
Twb
SATURATION LINE
FIG. 9
RECIRC CYCLE
OPTION-3
W, HUMIDITY RATIO, LB. MOISTURE / LB. DRY AIR
50% RH
7
6
HEAT TRANSFER
(TYP.) 4 PLACES
2
3
AMBIENT
1
4
5
HEAT ADDED
ROOM
8
11
10
9
75°
90°
TDB
145°
190°
TEMPERATURE, DRY BULB, °F

TABLE 1

RESULTS FOR SET A

Ventilation Mode

| $A_c$ ($m^2$) | Qsolar ($10^6$W-hr) | Auxiliary ($10^6$W-hr) | COP | Percent Solar | Hours of operation |
|---|---|---|---|---|---|
| 135 | 19.78 | 5.11 | 0.13 | 79 | 672 |
| 100 | 13.80 | 1.68 | 0.21 | 89 | 611 |
| 75 | 9.73 | 0.96 | 0.30 | 91 | 615 |
| 40 | 4.74 | 1.51 | 0.30 | 76 | 847 |

Recirculation Mode, Option 1

| $A_c$ | Qsolar | Auxiliary | COP | % Solar | Hours |
|---|---|---|---|---|---|
| 135 | 21.12 | 0.98 | 0.15 | 96 | 440 |
| 100 | 15.06 | 0.72 | 0.21 | 95 | 444 |
| 75 | 11.21 | 0.73 | 0.27 | 94 | 495 |
| 40 | 5.88 | 1.90 | 0.40 | 76 | 763 |

Recirculation Mode, Option 3

| $A_c$ | Qsolar | Auxiliary | COP | % Solar | Hours |
|---|---|---|---|---|---|
| 135 | 9.99 | 0.16 | 0.33 | 98 | 365 |
| 100 | 8.35 | 0.14 | 0.33 | 98 | 386 |
| 75 | 7.25 | 0.18 | 0.44 | 98 | 440 |
| 40 | 5.15 | 1.37 | 0.48 | 79 | 728 |

TABLE 2

RESULTS FOR SETS B AND C

Ventilation Mode

| $A_c$ $(M^2)$ | Flow Rate (cfm) | Qsolar $(10^6 W\text{-}hr)$ | Auxiliary $(10^6 W\text{-}hr)$ | COP | Percent solar | Hours of operation | Water Used (kg) |
|---|---|---|---|---|---|---|---|
| 75 | 1540 | 10.12 | 0.45 | 0.31 | 96 | 561 | 3620 |
| 55 | 1540 | 8.32 | 1.36 | 0.33 | 86 | 646 | 4238 |
| 40 | 1540 | 6.54 | 2.65 | 0.34 | 71 | 743 | 4759 |
| 25 | 1540 | 3.90 | 4.96 | 0.35 | 44 | 836 | 5045 |
| 55 | 1680 | 8.70 | 1.81 | 0.30 | 83 | 666 | 5173 |
| 55 | 1400 | 7.94 | 0.98 | 0.36 | 89 | 637 | 3353 |
| 55 | 1260 | 7.52 | 0.70 | 0.39 | 91 | 632 | 2594 |

Recirculation Mode, Option 1

| $A_c$ | Flow | Qsolar | Auxiliary | COP | % Solar | Hours | Water |
|---|---|---|---|---|---|---|---|
| 75 | 1540 | 11.84 | 0.54 | 0.26 | 96 | 484 | 4614 |
| 55 | 1540 | 9.68 | 1.00 | 0.30 | 91 | 526 | 4668 |
| 40 | 1540 | 7.70 | 1.94 | 0.33 | 80 | 584 | 4710 |
| 25 | 1540 | 4.93 | 4.22 | 0.35 | 54 | 640 | 4736 |
| 55 | 1680 | 9.70 | 1.18 | 0.29 | 87 | 520 | 4748 |
| 55 | 1400 | 9.37 | 0.86 | 0.32 | 92 | 540 | 4577 |
| 55 | 1260 | 9.01 | 0.80 | 0.33 | 92 | 559 | 4520 |

TABLE 2 (CONT.)

Recirculation Mode, Option 2

| $A_c$ | Flow | Qsolar | Auxiliary | COP | % Solar | Hours | Water* |
|---|---|---|---|---|---|---|---|
| 75 | 1540 | 10.39 | 0.28 | 0.31 | 97 | 371 | 1776 |
| 55 | 1540 | 8.55 | 0.41 | 0.37 | 95 | 394 | 1743 |
| 40 | 1540 | 7.02 | 0.64 | 0.42 | 92 | 432 | 1714 |
| 25 | 1540 | 4.77 | 1.94 | 0.48 | 71 | 483 | 1629 |
| 55 | 1680 | 8.95 | 0.40 | 0.35 | 96 | 386 | 1756 |
| 55 | 1400 | 8.30 | 0.32 | 0.38 | 96 | 405 | 1737 |
| 55 | 1260 | 8.30 | 0.24 | 0.40 | 97 | 423 | 1750 |

*Does not include water use of indirect evaporative cooler.

Recirculation Mode, Option 3

| $A_c$ | Flow | Qsolar | Auxiliary | COP | % Solar | Hours | Water |
|---|---|---|---|---|---|---|---|
| 75 | 1540 | 7.25 | 0.17 | 0.43 | 98 | 436 | 4484 |
| 55 | 1540 | 6.87 | 0.26 | 0.46 | 96 | 468 | 4569 |
| 40 | 1540 | 6.08 | 0.66 | 0.48 | 90 | 520 | 4581 |
| 25 | 1540 | 4.22 | 2.20 | 0.49 | 66 | 610 | 4639 |
| 55 | 1680 | 7.03 | 0.20 | 0.45 | 97 | 451 | 4573 |
| 55 | 1400 | 6.76 | 0.26 | 0.47 | 96 | 486 | 4490 |
| 55 | 1260 | 6.66 | 0.27 | 0.47 | 96 | 512 | 4444 |

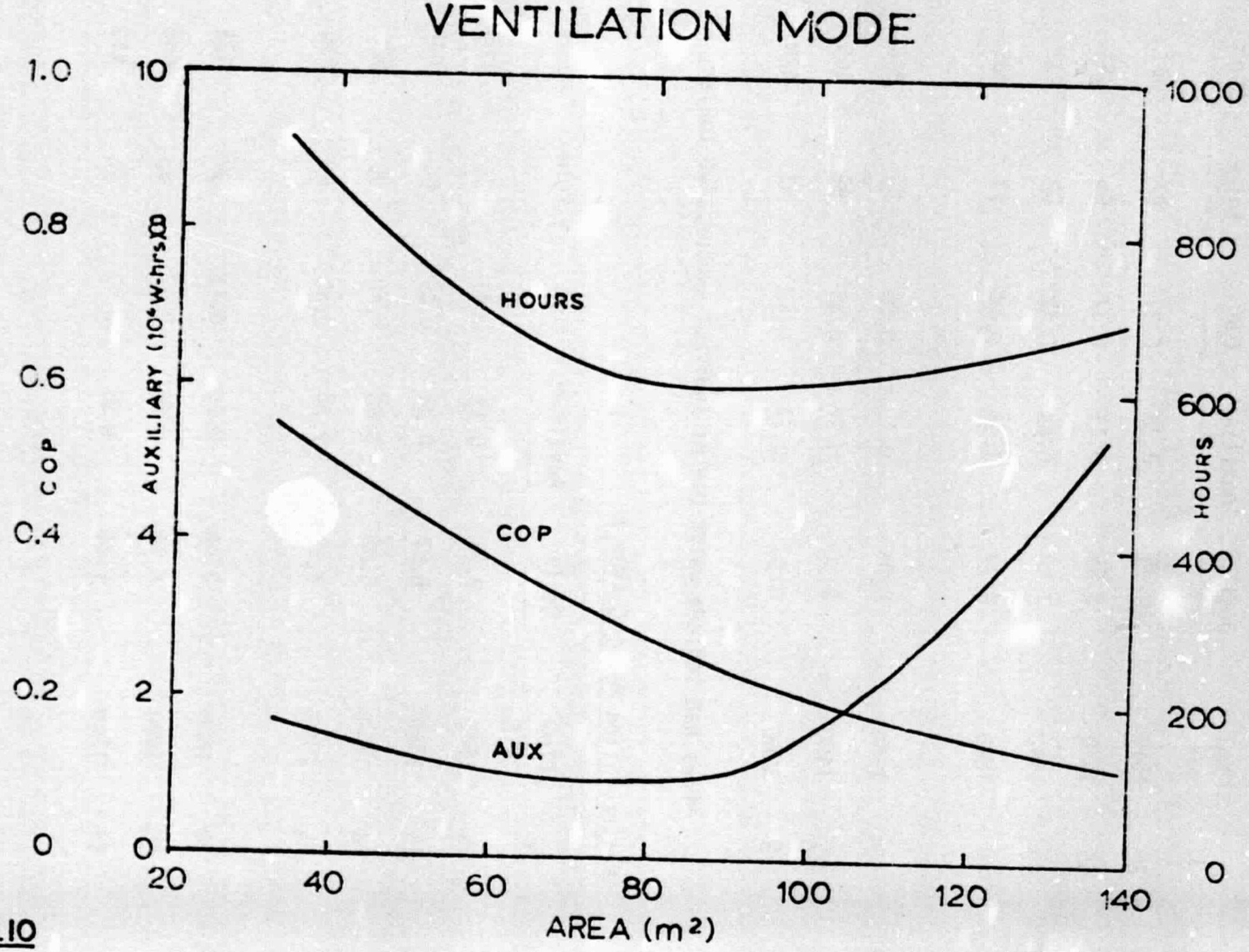

VENTILATION MODE
COP
0
0.2
0.4
0.6
0.8
1.0
AUXILIARY ($10^6$ W-hrs)
0
2
4
6
8
10
HOURS
0
200
400
600
800
1000
HOURS
COP
AUX
20
40
60
80
100
120
140
AREA ($m^2$)

FIG.10

# RECIRCULATION MODE OPTION 1

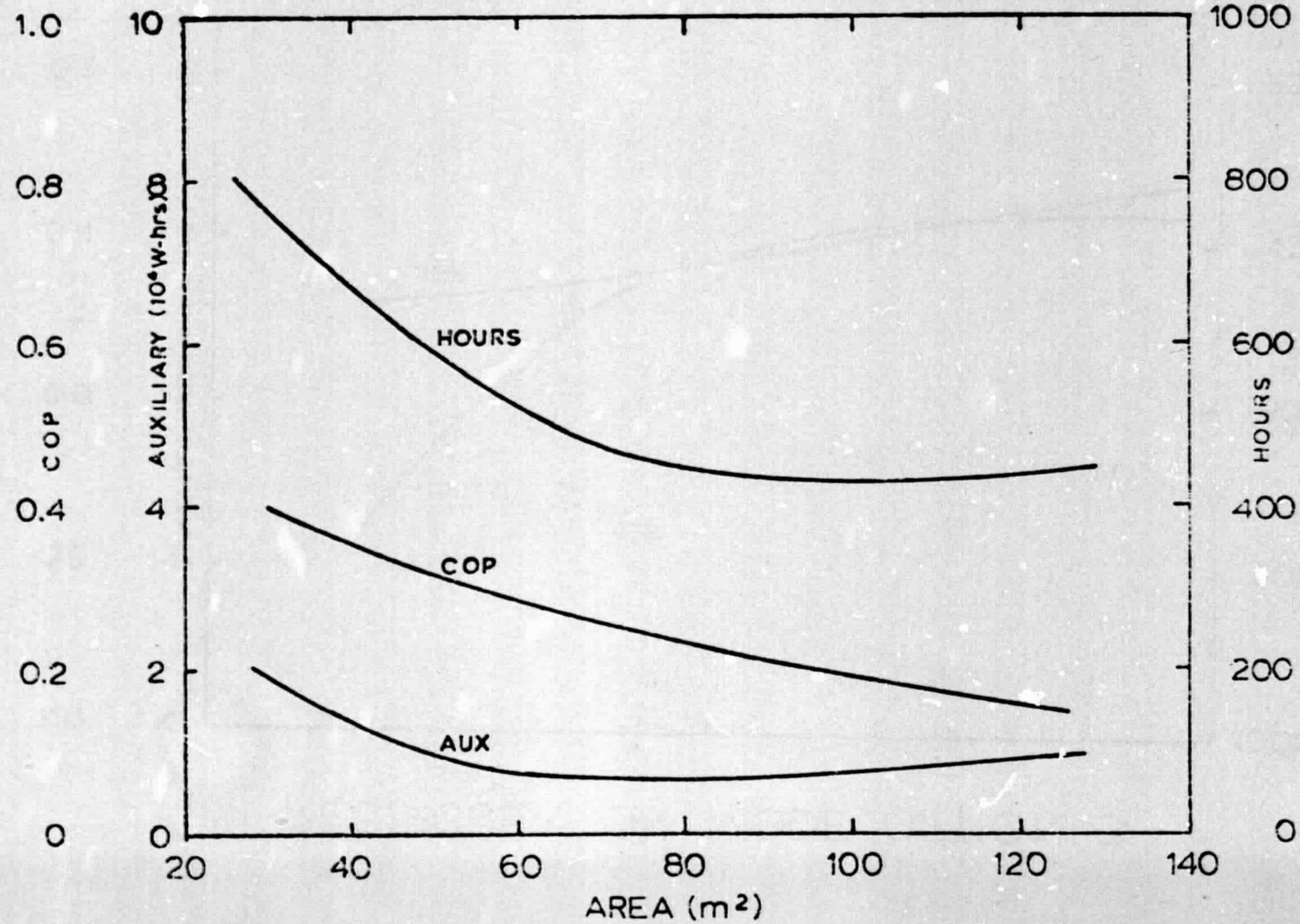



FIG. 11

# RECIRCULATION MODE OPTION 3

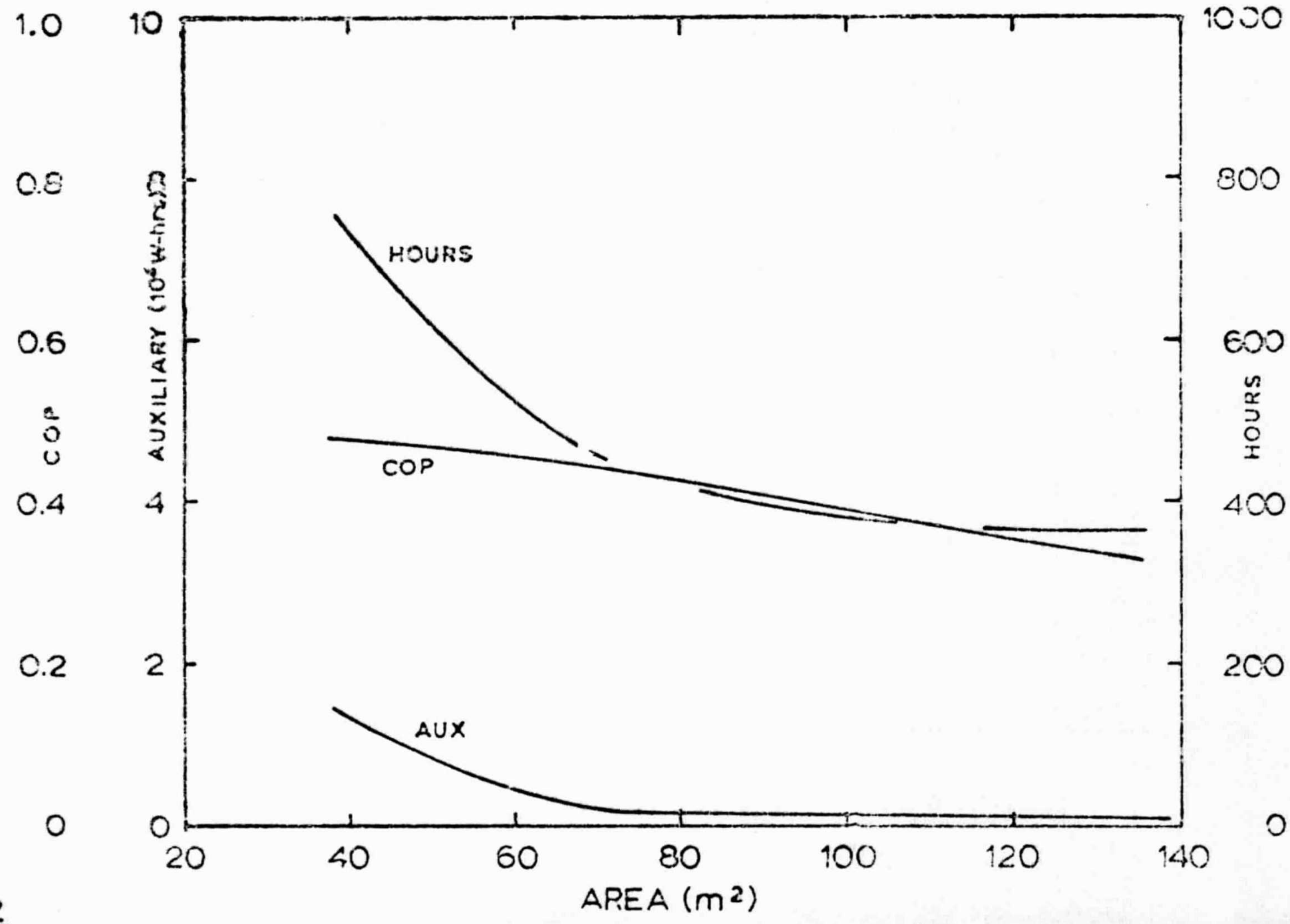



FIG. 12

# VENTILATION MODE

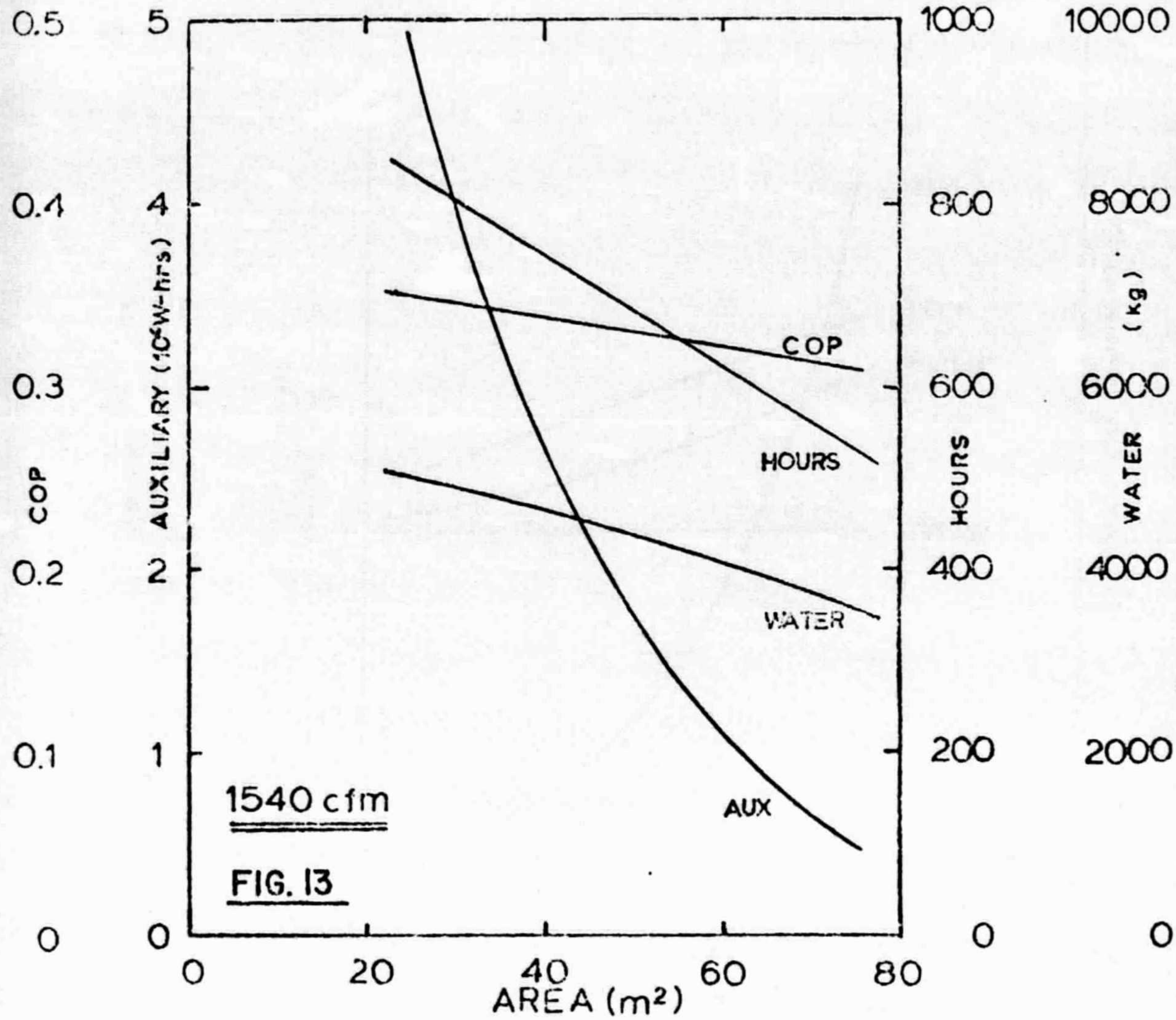



FIG. 13

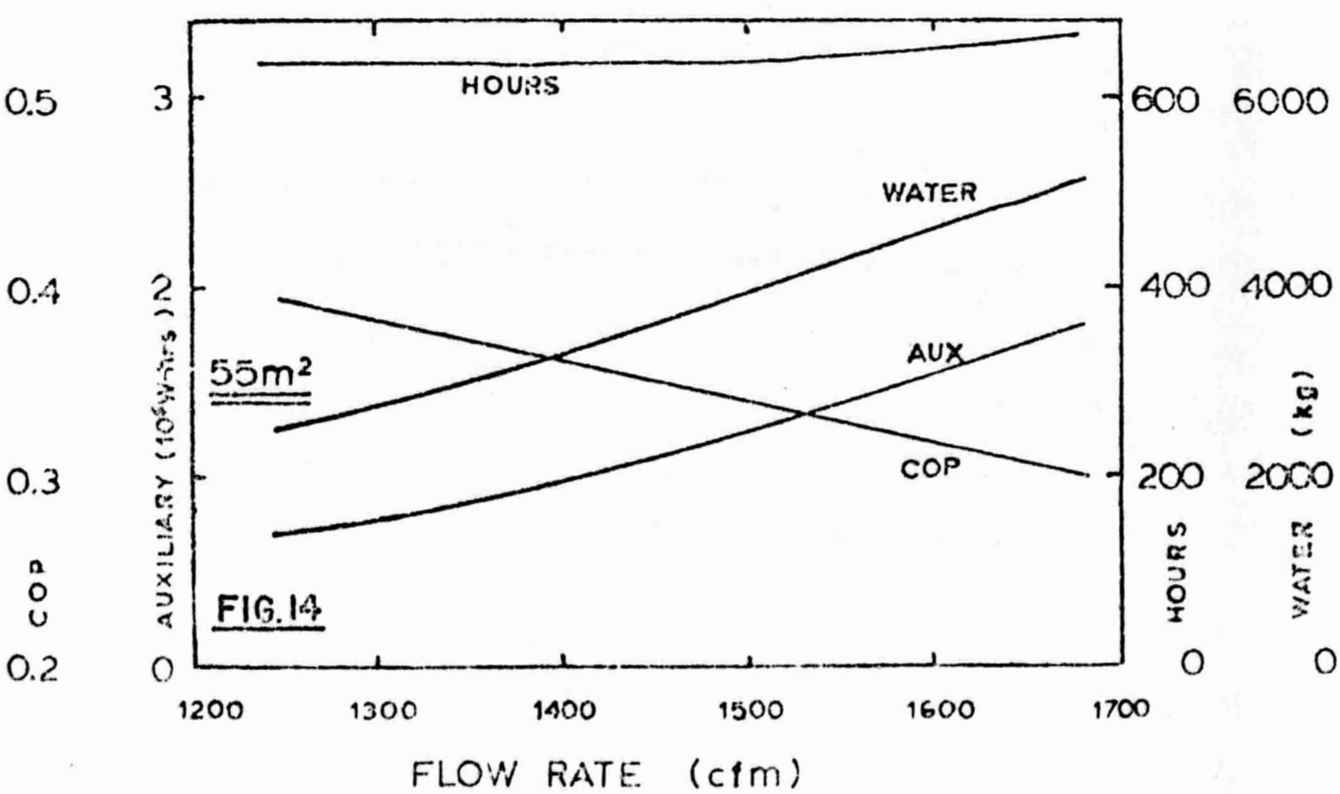
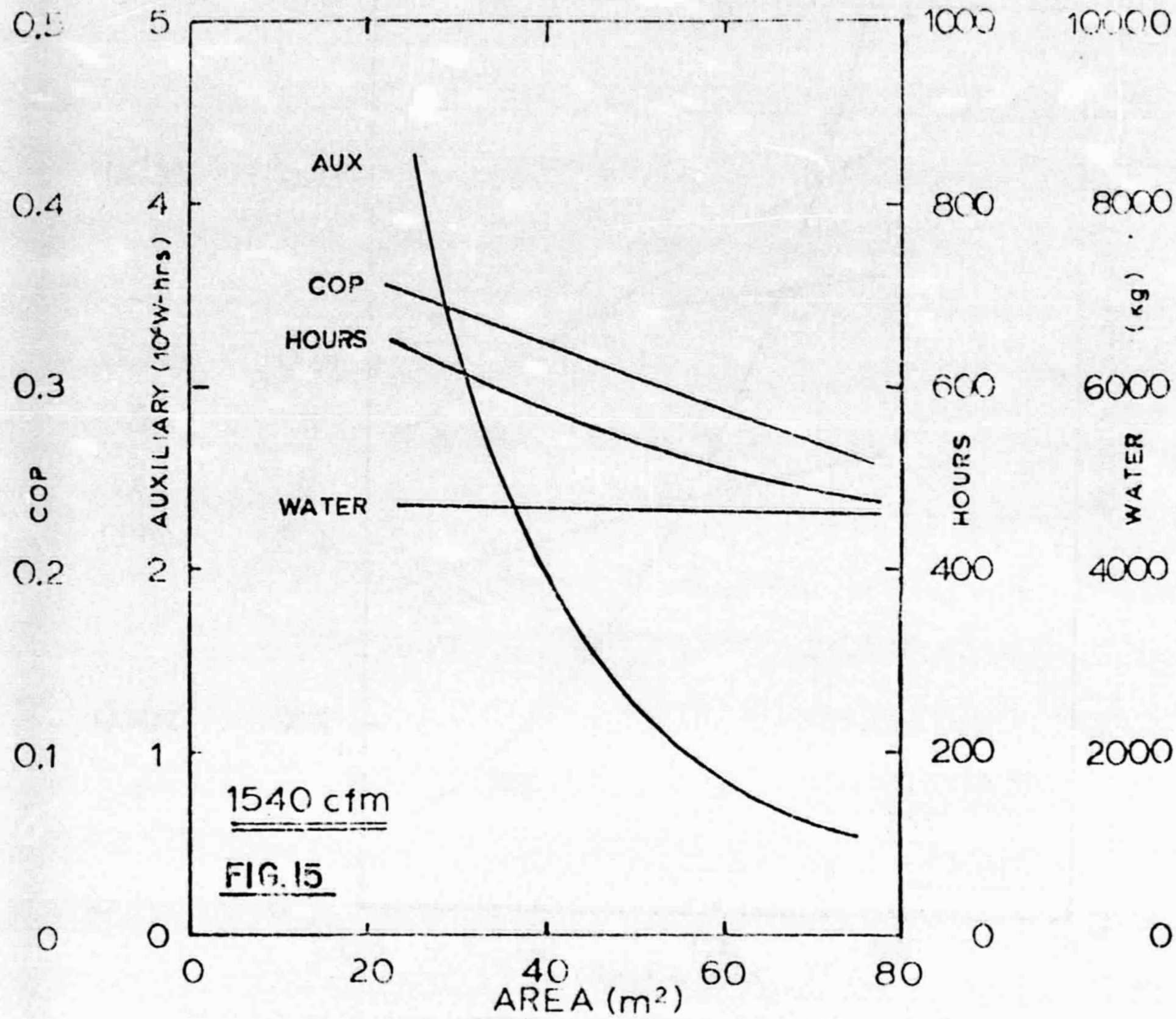
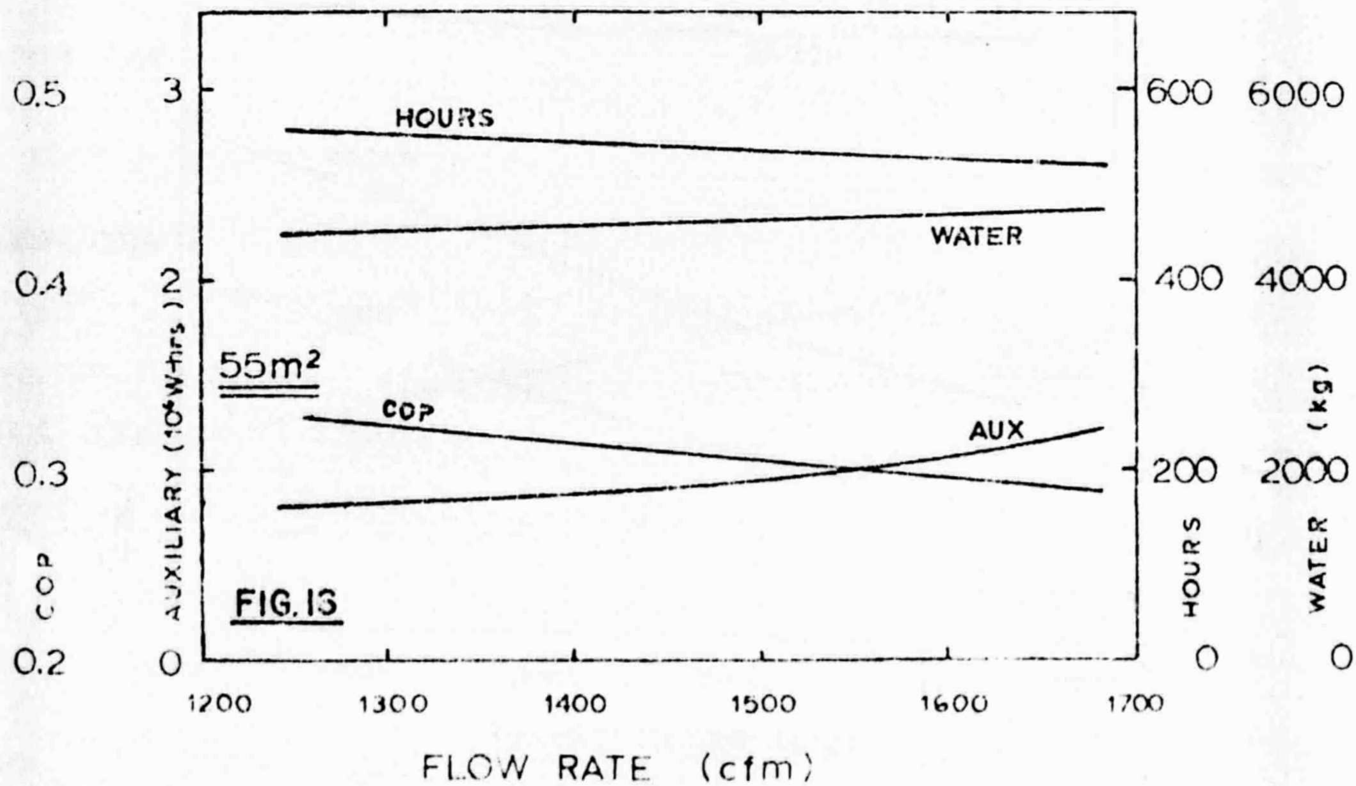
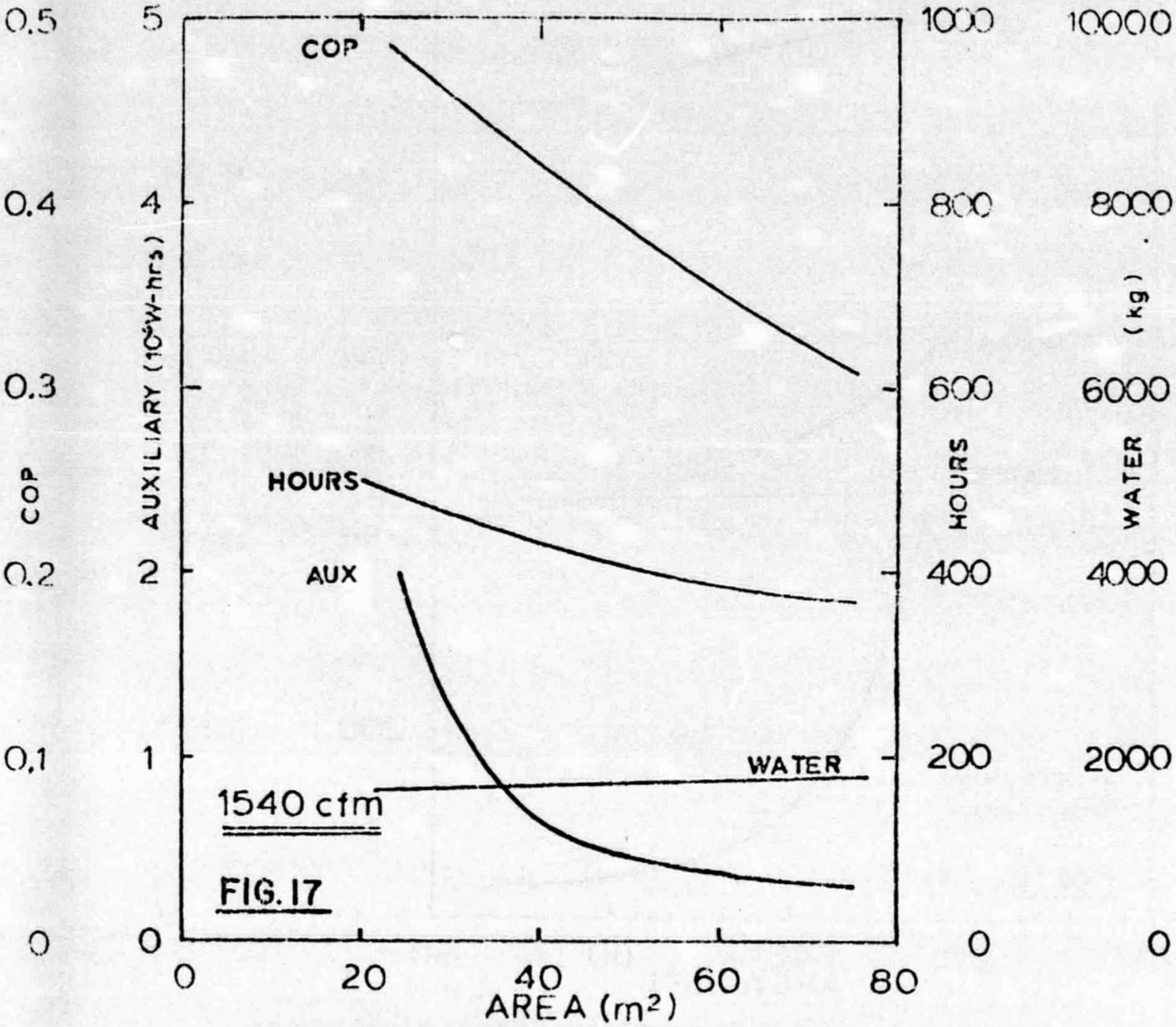


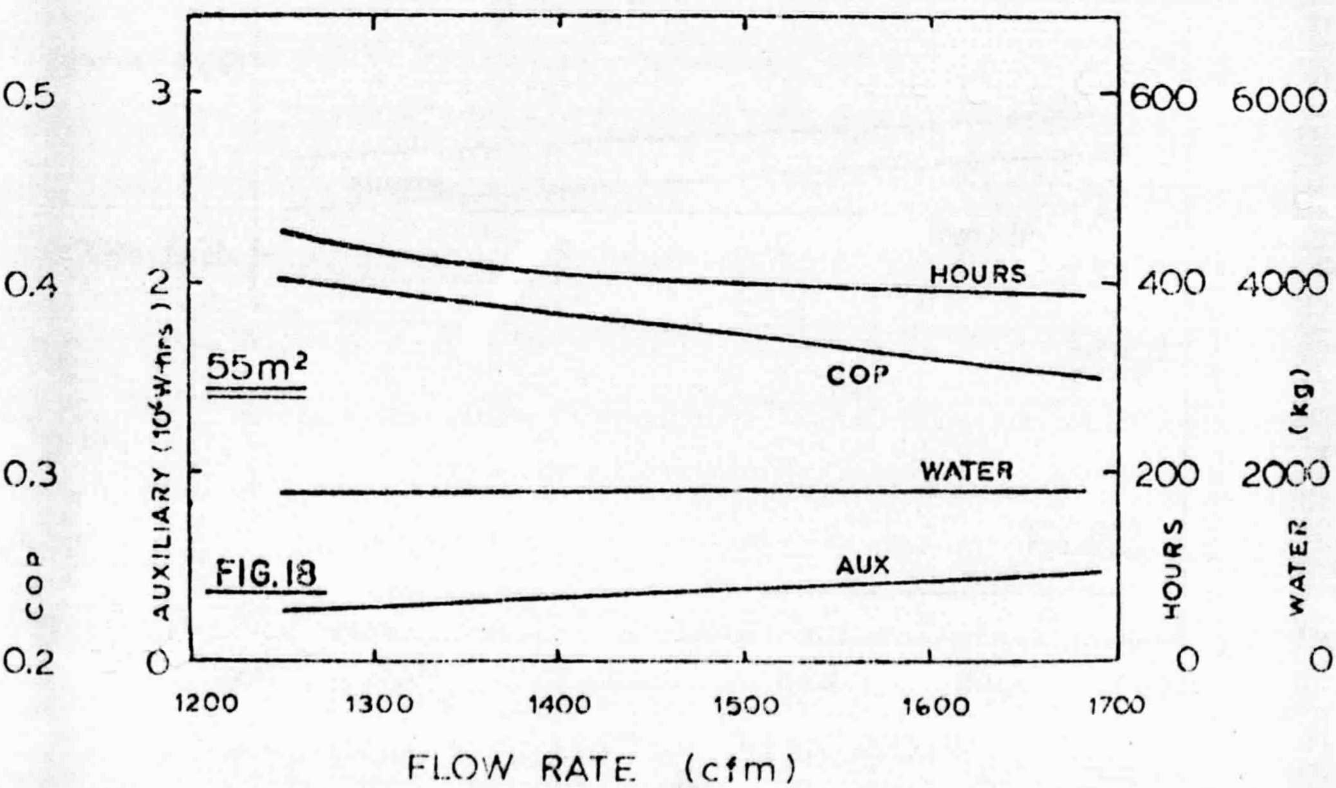

FIG. 14

# RECIRCULATION MODE OPTION 1

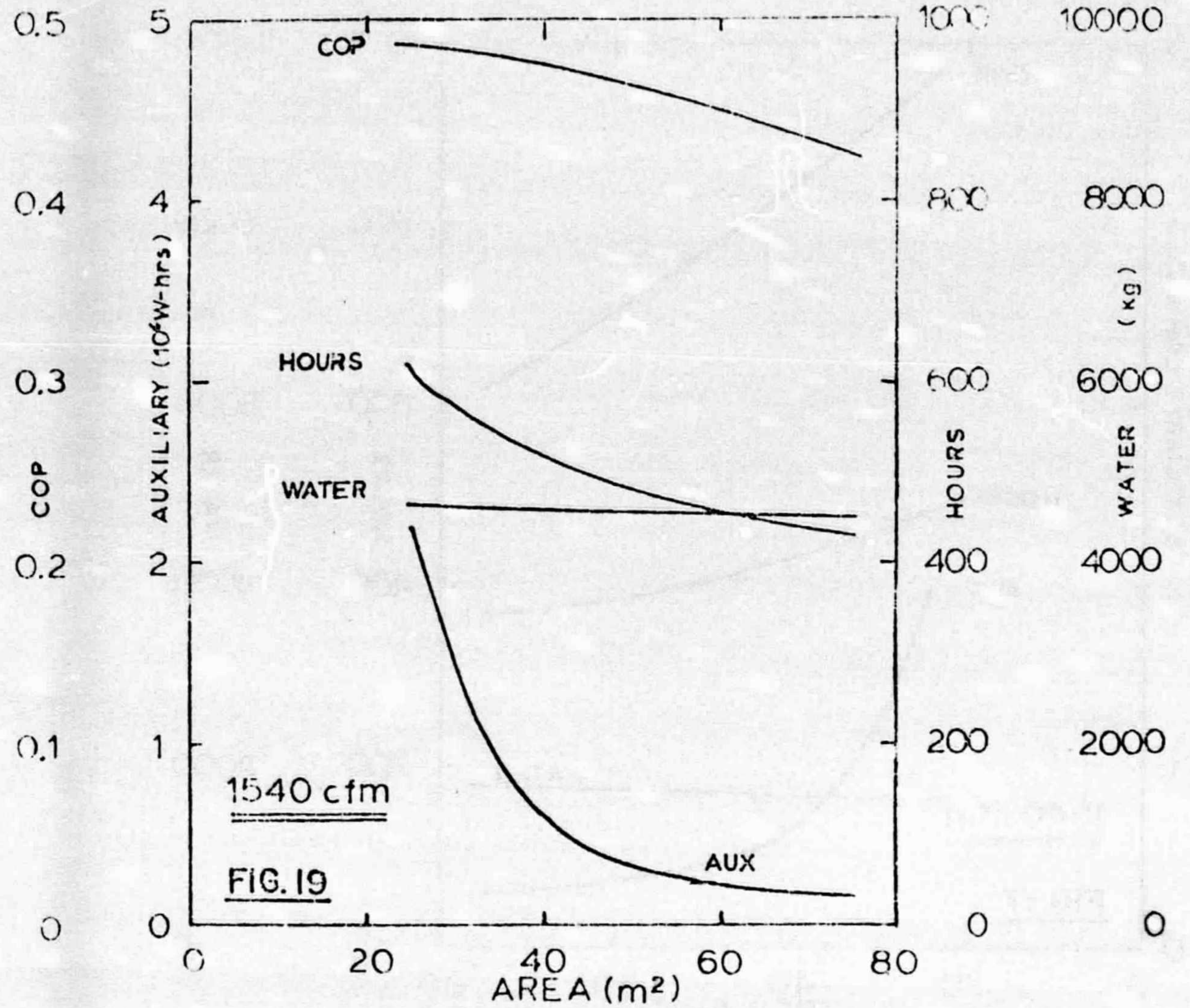


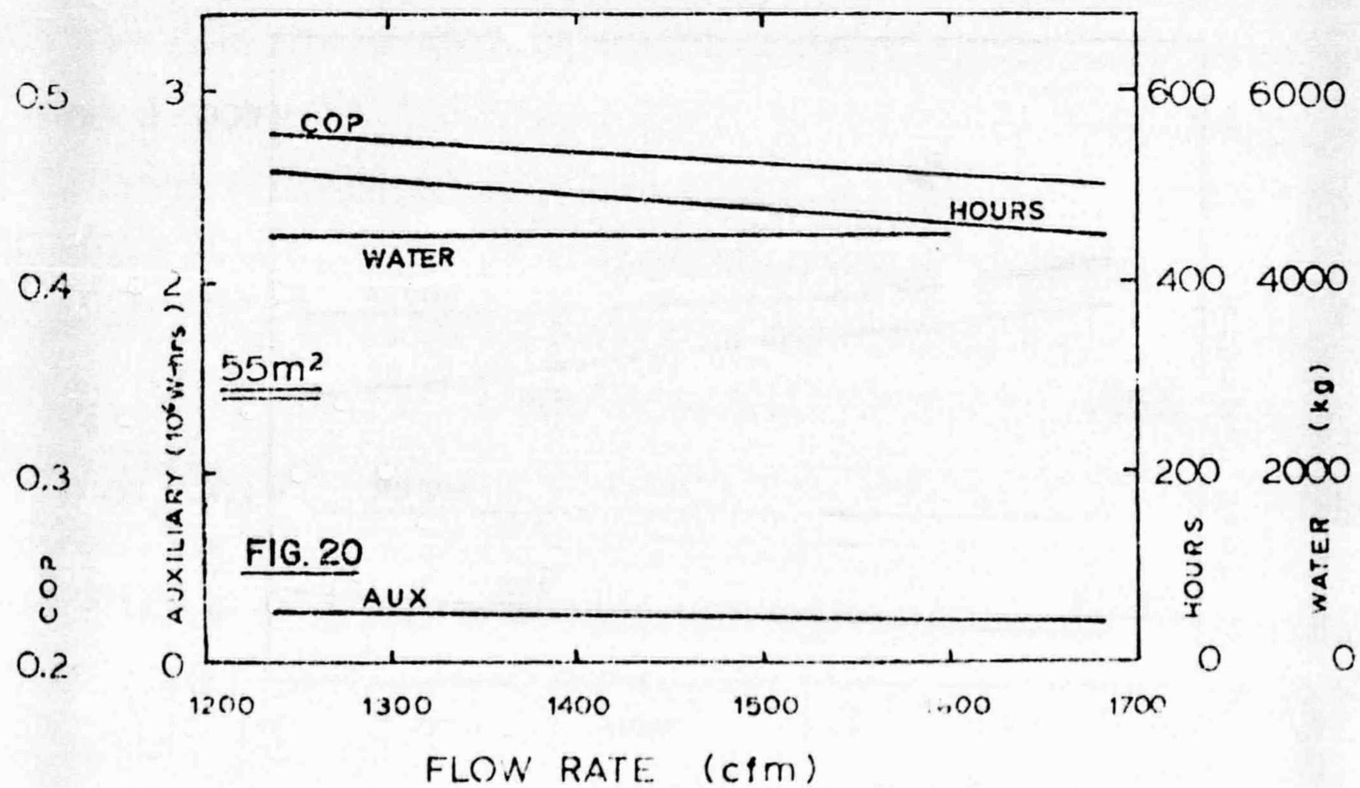




COP
HOURS
AUX
WATER
1540 cfm
FIG. 17
COP
AUXILIARY ($10^6$ W-hrs)
HOURS
WATER (kg)
AREA ($m^2$)
0.5 0.4 0.3 0.2 0.1 0
5 4 2 1 0
1000 800 600 400 200 0
10000 8000 6000 4000 2000 0
0 20 40 60 80


HOURS
COP
WATER
AUX
55m²
FIG. 18
COP
AUXILIARY ($10^6$ W-hrs)
HOURS
WATER (kg)
0.5 0.4 0.3 0.2
3 2 0
600 400 200 0
6000 4000 2000 0
1200 1300 1400 1500 1600 1700
FLOW RATE (cfm)

# RECIRCULATION MODE OPTION 3



SOLARON DESICCANT RECIRCULATION CYCLE
HEATING AND COOLING SYSTEM

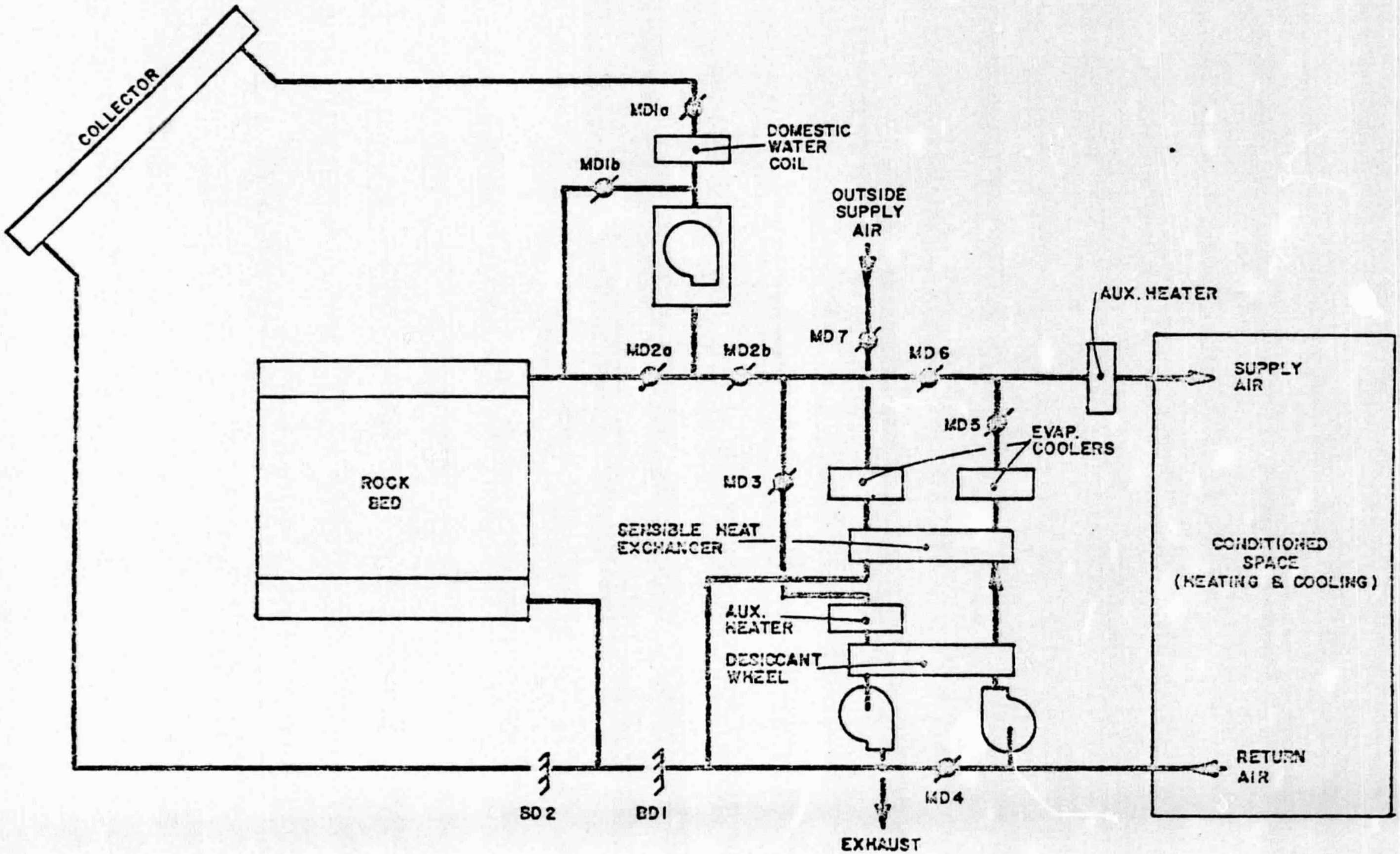



FIG. 21

ATTACHEMENT III

Handouts for ERDA Contractors Meeting

August 8-10, 1977

## OBJECTIVES:

(NASA CONTRACT - NAS3 - 32249)

1. TO DEVELOP A SOLAR ASSISTED HEATING AND COOLING SYSTEM, INCLUDING DOMESTIC HOT WATER.

2. PRODUCE, DELIVER, AND INSTALL TWO PROTOTYPE SYSTEMS IN SELECTED SINGLE FAMILY RESIDENCES.

3. MONITOR AND EVALUATE PERFORMANCE OF THESE SYSTEMS OVER A PERIOD OF ONE YEAR.

## MAJOR TASKS:

1. DEVELOP AN EFFECTIVE AND EFFICIENT SOLAR REGENERATED DESICANT COOLING SYSTEM.

2. SELECT SYSTEM HARDWARE COMPONENTS THAT ARE COMMERCIALLY AVAILABLE ON THE OPEN MARKET TO INSURE AVAILABILITY AND RELIABILITY.

3. OPTIMIZE THE SYSTEM, INCLUDING CONTROL LOGIC, USING THE "TRNSYS" COMPUTER SIMULATION PROGRAM, SUBSCALE AND PROTOTYPE TESTING.

4. INTEGRATE THE HEATING AND COOLING SYSTEMS, INCLUDING CONTROLS, TO REALIZE THE BEST EFFECTIVENESS OF EACH.

5. PRODUCE AND INSTALL A SYSTEM THAT WILL OPERATE RELIABLY.

SOLARON

## KEY PROBLEMS ADDRESSED:

1. SELECTION OF PSYCHROMETRIC CYCLE TO BE USED.——
   SELECTED DESICCANT DRYING/EVAPORATIVE COOLING CYCLE USING SILICA GEL AS THE DRYING AGENT AND MUNTERS MEDIA AS THE EVAPORATIVE COOLING SURFACE.

2. CYCLE HEAT EXCHANGE.——
   SELECTED AIR-TO-AIR PLATE TYPE HEAT EXCHANGER.

3. SOLAR ENERGY INPUT TO SYSTEM.——
   SELECTED STANDARD SOLARON AIR HEATING SOLAR COLLECTORS.

4. SELECTION OF CYCLE - VENT. (100% OUTSIDE AIR) - VS.- RECIRC. (100% RECIRCULATION).——
   SELECTION DEPENDS ON LOCAL CLIMATIC CONDITIONS. HAVE SELECTED RECIRC. CYCLE FOR SITE IN AKRON, OHIO.

## TECHNICAL ACCOMPLISHMENTS:

BASED ON THE WORK DONE BY D.J. CLOSE, I.L. McCLAINS-CROSS AND P.J. BANKS IN AUSTRALIA, THE UNIVERSITY OF WISCONSIN HAS DEVELOPED A COMPUTER SIMULATION PROGRAM FOR ANALYZING SOLAR/DESICCANT (SILICA-GEL) COOLING SYSTEMS. THE PROGRAM HAS BEEN CROSS-CHECKED AGAINST ACTUAL TEST DATA GENERATED BY DR. CLOSE AND IS IN CLOSE AGREEMENT. SOLARON IS USING THE WISCONSIN SIMULATION PROGRAM UNDER SUB-CONTRACT TO ASSIST IN THE DEVELOPMENT OF A SOLAR/DESICCANT COOLING SYSTEM UNDER NASA CONTRACT. ALSO, SOLARON IN COOPERATION WITH BRY-AIR, WILL BE GENERATING MORE TEST DATA ON THE ACTUAL DESICCANT WHICH WILL BE USED, AND THIS DATA WILL BE GIVEN TO WISCONSIN TO FURTHER FINE TUNE THE MODEL.

## CONCLUSIONS & RECOMMENDATIONS:

NONE AT THIS POINT-OF-TIME IN THE CONTRACT.

## FUTURE ACTIVITIES:

THE DEVELOPMENT OF A SUCCESSFUL SOLAR/DESICCANT COOLING SYSTEM LOOKS VERY PROMISING. FUTURE EFFORT SHOULD BE DIRECTED TOWARD REDUCING THE COST OF HARDWARE COMPONENTS. CURRENTLY ONLY LARGE, HEAVY INDUSTRIAL EQUIPMENT IS AVAILABLE ON THE MARKET. THIS EQUIPMENT IS OVER-KILL FOR THE RESIDENTIAL AND MOST OF THE COMMERCIAL MARKET.

## NARRATIVE DESCRIPTION OF FIGURES

Fig. 1 This figure is a schematic of a Solaron heating system integrated with any of the proposed desiccant air conditioning systems. The basic modes of operation of the system allow the desiccant system to be regenerated directly from the collector or from the stored heat in the pebble bed. The system can also store heat from the collector in the pebble bed, for periods when there is solar collector heat available and there is no simultaneous need for cooling.

Fig's. 2 through 9 This group of figures shows the variety of solar assisted desiccant cooling system modes presently under study and development by Solaron Corporation. Each mode is described by a system flow schematic (Fig's. 2, 4, 6 & 8) followed by a corresponding psychrometric cycle diagram (Fig's. 3, 5, 7 & 9).

Fig's. 2 & 3 Vent Cycle - This proposed cycle is called a vent cycle, due to its use of 100% fresh air. Referring to the system flow diagram and psychrometric chart, ambient air at state 1 is drawn through a desiccant wheel, where it is dried and heated to state 2. The air then is cooled by a sensible air-to-air heat exchanger to state 3 followed by evaporative cooling to state 4, where it enters the room. Return air from the room (state 5) is first evaporatively cooled to 6 and then is heated by the sensible heat exchange process to state 7. At this point it is further heated by the solar system (either by the collector or storage, Fig. 1) and an auxiliary heater if needed, to state 8. This hot air then regenerates the desiccant wheel and is discharged to the atmosphere at state 9.

Fig's. 4 & 5 Recirculation Cycle, Option 1 - This is one of three proposed recirculation cycles. Recirculation in this case refers to the room air stream flowing in a closed loop, while a second open loop air stream is used for regeneration. Referring to the room air loop first, return air at 6 is drawn through a desiccant wheel where it is dried and heated to 7; the hot dry air at state 7 is then sensibly cooled to state 8 and evaporatively cooled to state 9 where it re-enters the rooms. The regeneration cycle consists of ambient air at state 1 which is evaporatively cooled (state 2) and then sensibly heated (state 3). Air at state point 3 is then heated by the solar system and/or auxiliary (if required) to state 4; this air then is used to regenerate the desiccant wheel and is discharged to atmosphere at state 5.

Fig's. 6 & 7 Recirculation Cycle, option 2 - In this cycle a slight change is made to the system in option 1 in an attempt to increase the overall COP. Return air at state 5 is dried to state 6 and sensibly cooled as before, however at state 7 the air is sensibly cooled further by an indirect evaporative cooler, which allows state 8 to closely approach the outdoor wet bulb temperature. Air at state 8 is then evaporatively cooled to 9 where it enters the room. Ambient air used in the regenerative loop actually takes two paths. A relatively large flow is used in the indirect evaporative cooler to provide cooling to the room air stream, while an additional amount of air equal in flow to the room air rate is directed into the sensible heat exchanger. This air is heated to the same temperature at state 2 as was the air at state 3 of option 1, but since it has not been evaporatively cooled in this cycle it is at a lower humidity. This allows the air at state 3 of this cycle to be at a lower temperature and still achieve the required regeneration of the wheel.

Fig's. 8 & 9 Recirculation Cycle, option 3 - This cycle is exactly the same as option 1 on the room side, but is significantly different on the regeneration side. Specifically, ambient air enters at two different points in the regeneration process. One ambient air stream is evaporatively cooled from state 1 to 2 and then heated from 2 to 3 by the first sensible heat exchanger. Ambient air at state 1 also enters a second sensible heat exchanger and is heated to state 4; this air is then further heated by the solar system and/or auxiliary to state 5. This air then regenerates the wheel and leaves at state 6 where it enters the second sensible heat exchanger, is cooled and is discharged at state 7.

Fig's. 10 through 17, present graphically the preliminary results of a computerized cycle simulation program at the University of Wisconsin. This particular study was run on a specific house in New York City, N.Y. using hourly weather data recorded in the period of 1 July through 22 September 1957. In this way comparison of various cycle parameters for the modes and system cycles is directly related to the same set of operating conditions and loads. The parameters analyzed in this study are COP, hours of system operation, water consumption, and auxiliary energy usage, all totalized over the entire period of the study, 1 July through 22 September.

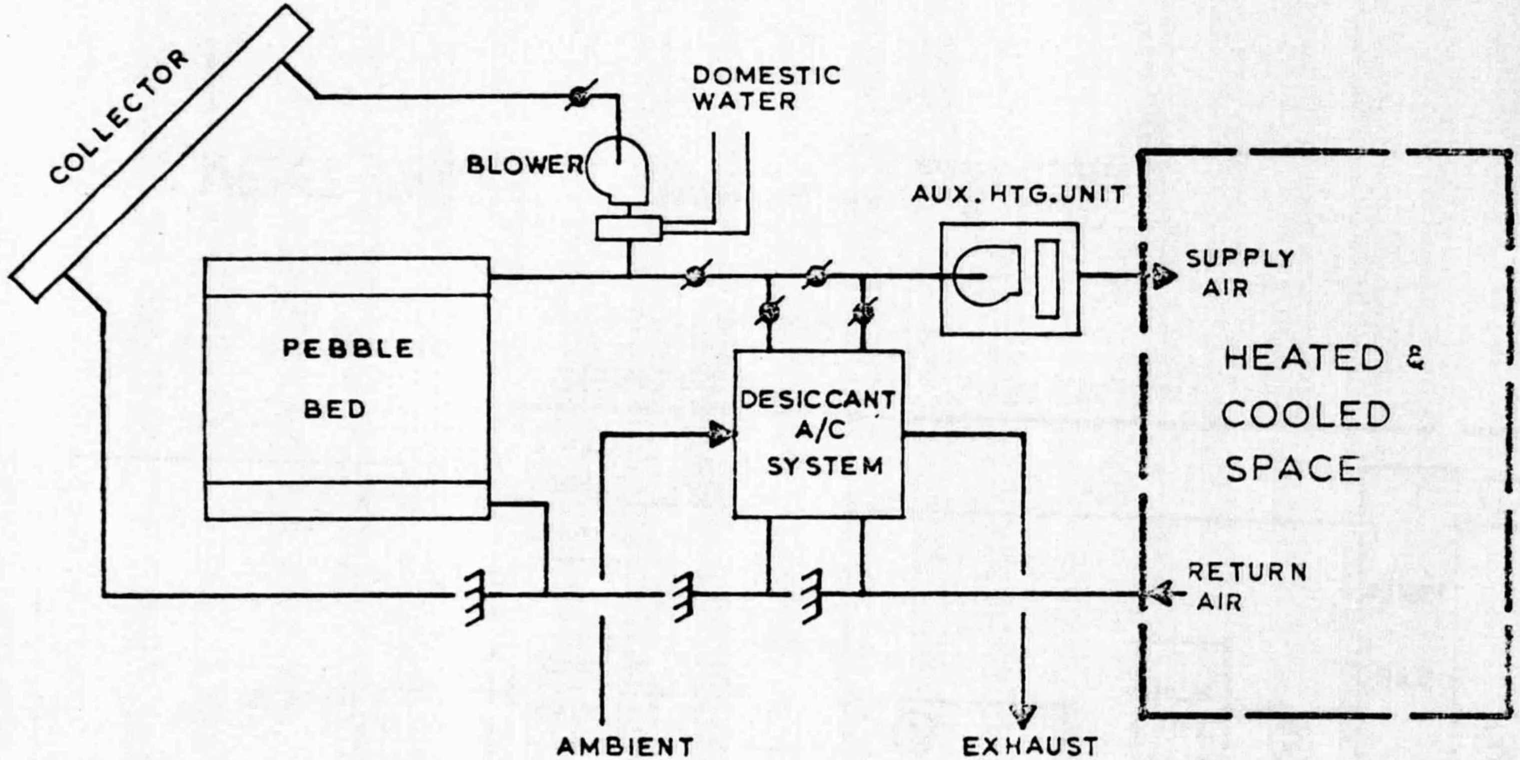
Fig's. 10, 12, 14 & 16, present the above data for a constant air flow rate of 1540 SCFM and a variable collector area from 25 to 75 square meters.

Fig's. 11, 13, 15 & 17, present the same data for a constant collector area of 55 square meters and a variable air flow rate from 1260 to 1680 SCFM.

Fig. 18 This is a schematic flow diagram for an integrated solar heating and cooling system with additional provision for domestic hot water. The cooling portion of this system is Solaron desiccant recirculation cycle, option 1, as shown in Fig's. 2 & 3.

COP - Coefficient of Performance (COP) as shown and used in these figures is based on the total energy input to the desiccant cycle, including both conventional and collected solar heat and is equal to total cycle cooling output divided by total heat input to the cycle. This parameter is useful in a comparative evaluation of the various cycles.

For evaluating systems relative to conventional energy usage, the COP should be evaluated as total system output divided by conventional energy input only. These values of COP will be generated in future analyses.



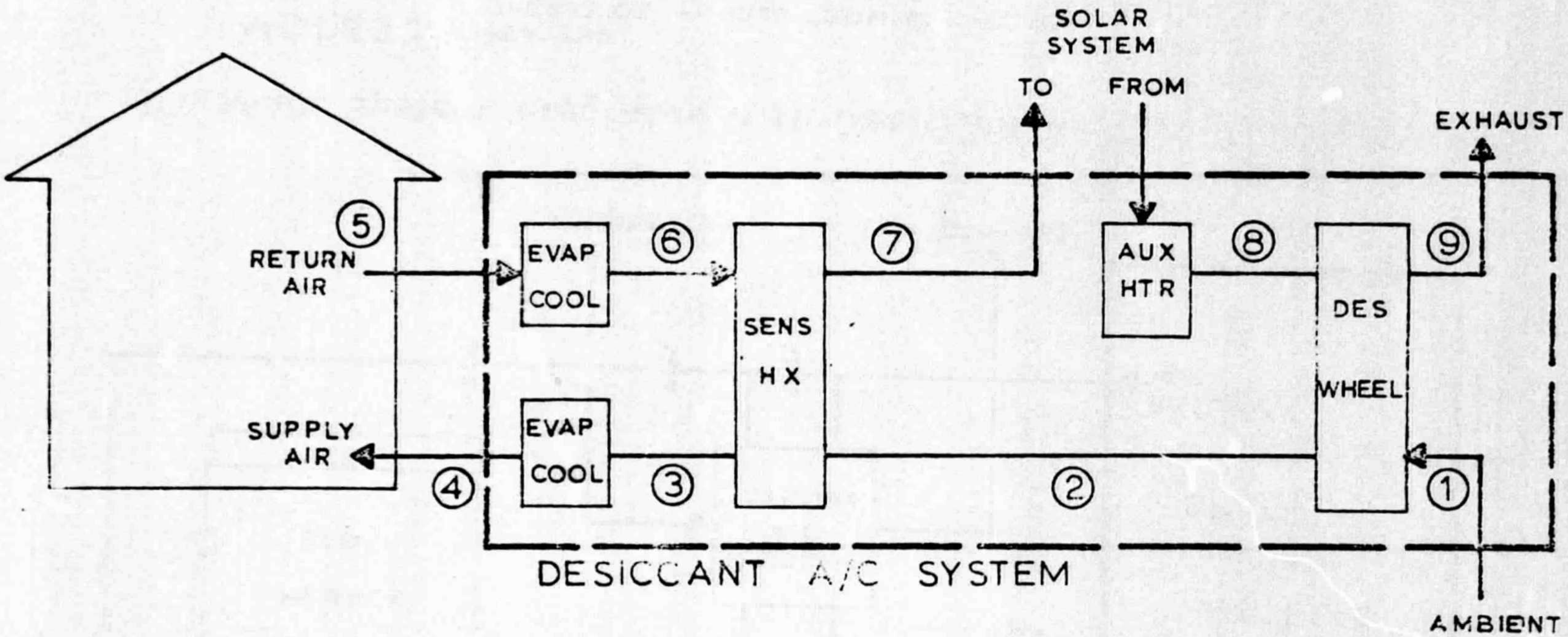

SOLARON System Integrated With Desiccant Air Conditioning System

MODES:

HEATING OPERATION
- HEAT FROM COLLECTOR
- STORE HEAT
- HEAT FROM STORAGE

COOLING OPERATION
- REGENERATE FROM COLLECTOR
- STORE HEAT
- REGENERATE FROM STORAGE

**FIG. 1**




VENT CYCLE

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

FIG. 2

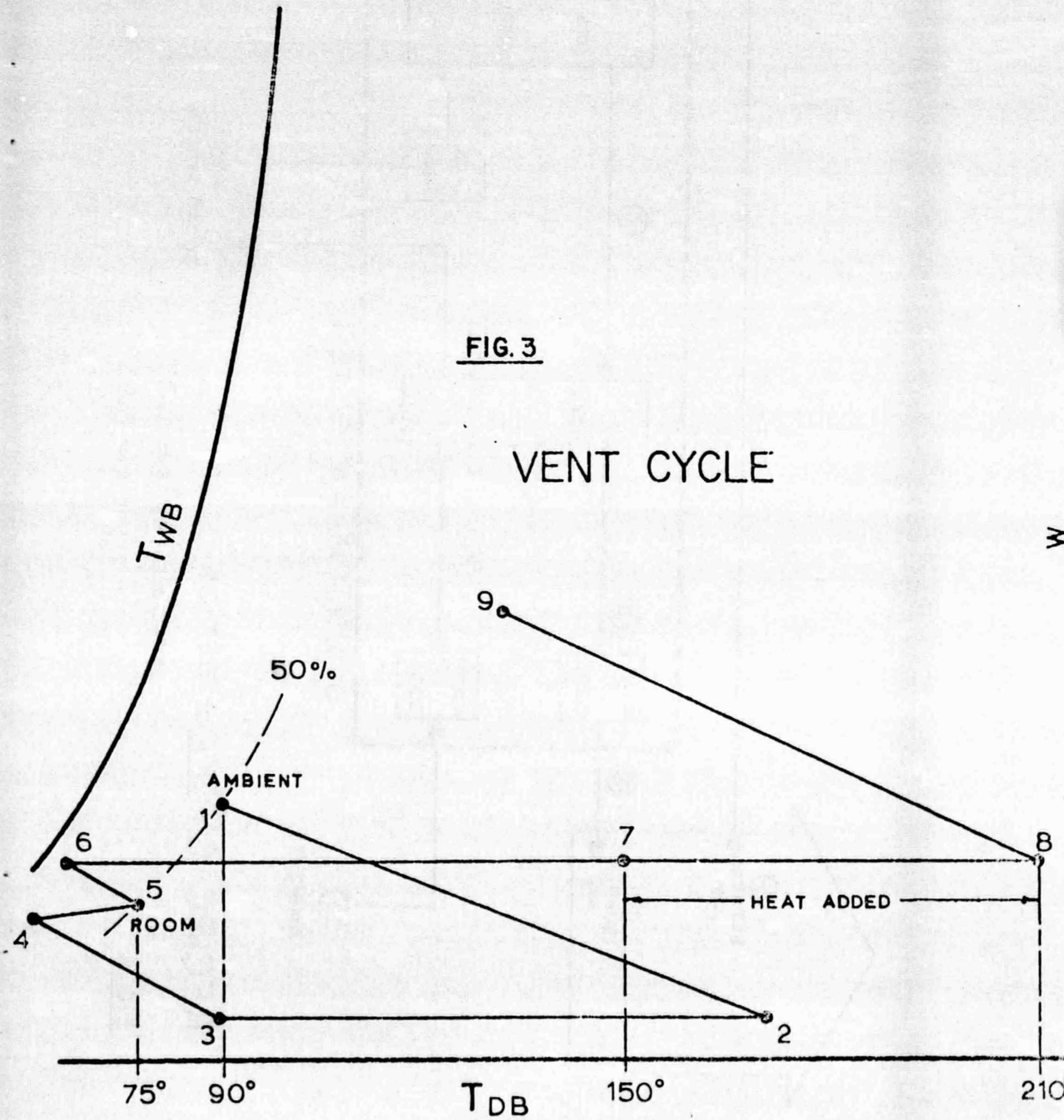
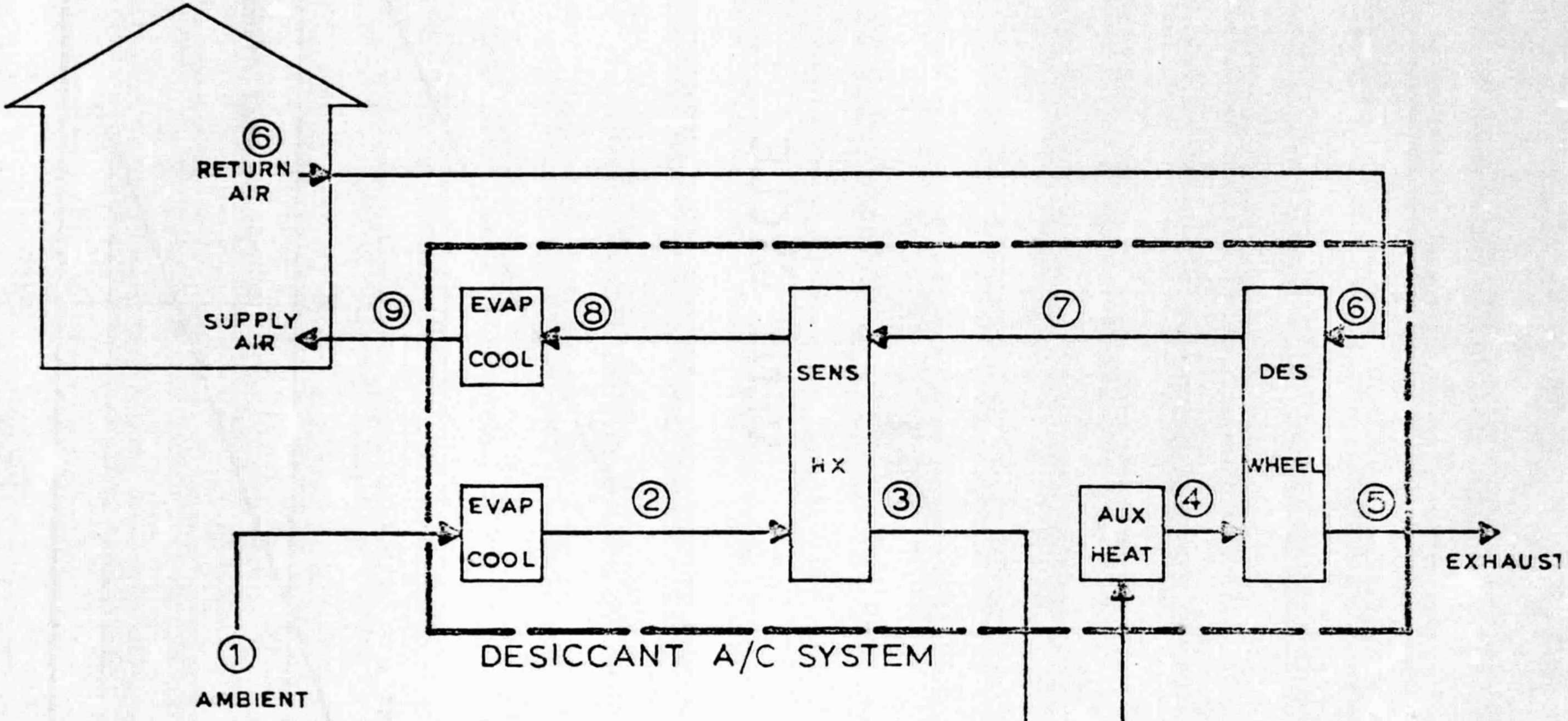


FIG. 3
VENT CYCLE
$T_{WB}$
W
50%
AMBIENT
1
2
3
4
5
ROOM
6
7
8
9
HEAT ADDED
75°
90°
150°
210
$T_{DB}$




RECIRC CYCLE - OPTION 1

SENSIBLE HX $\varepsilon = 0.70$

$\varepsilon = 0.85$

FIG. 4

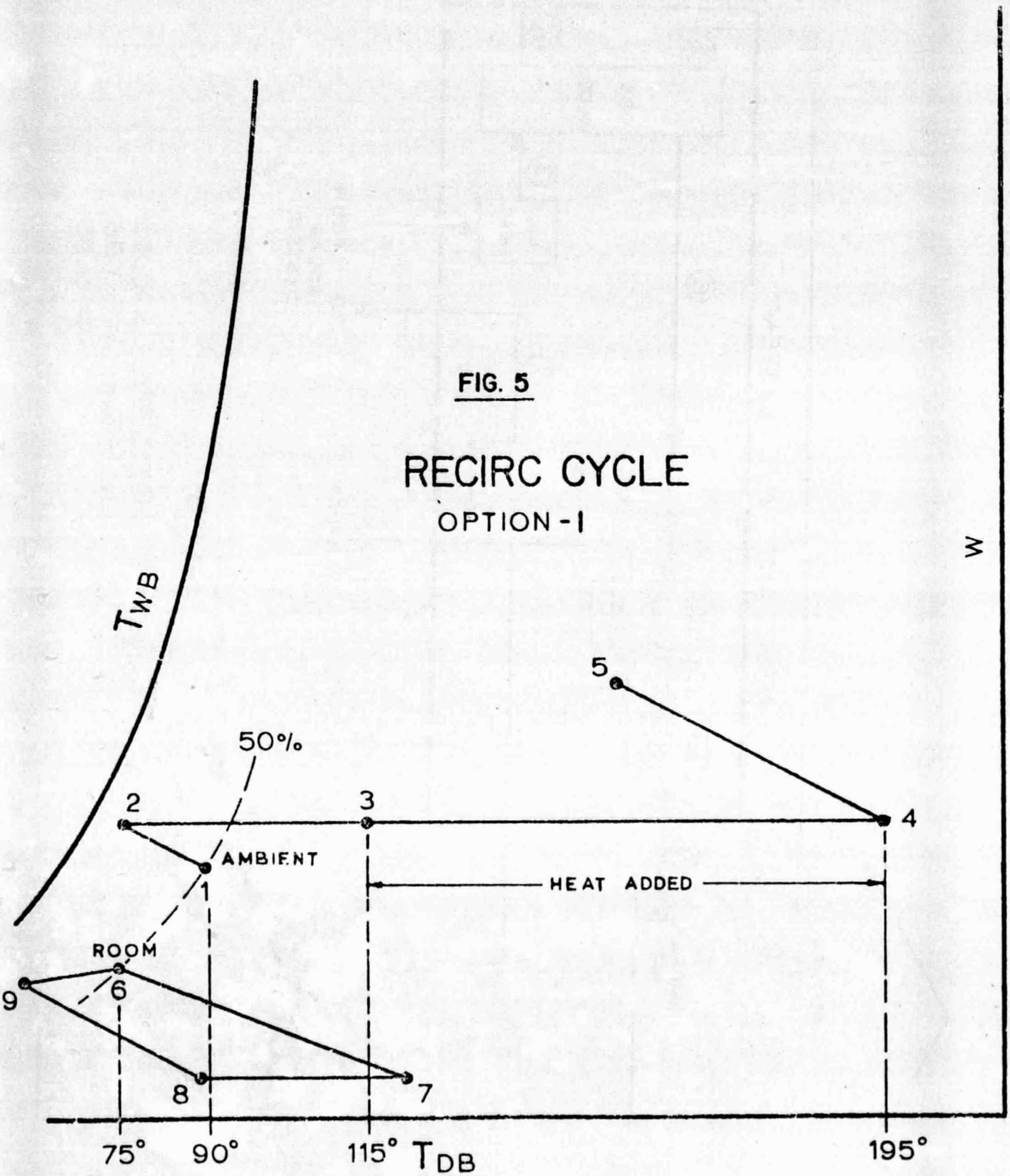
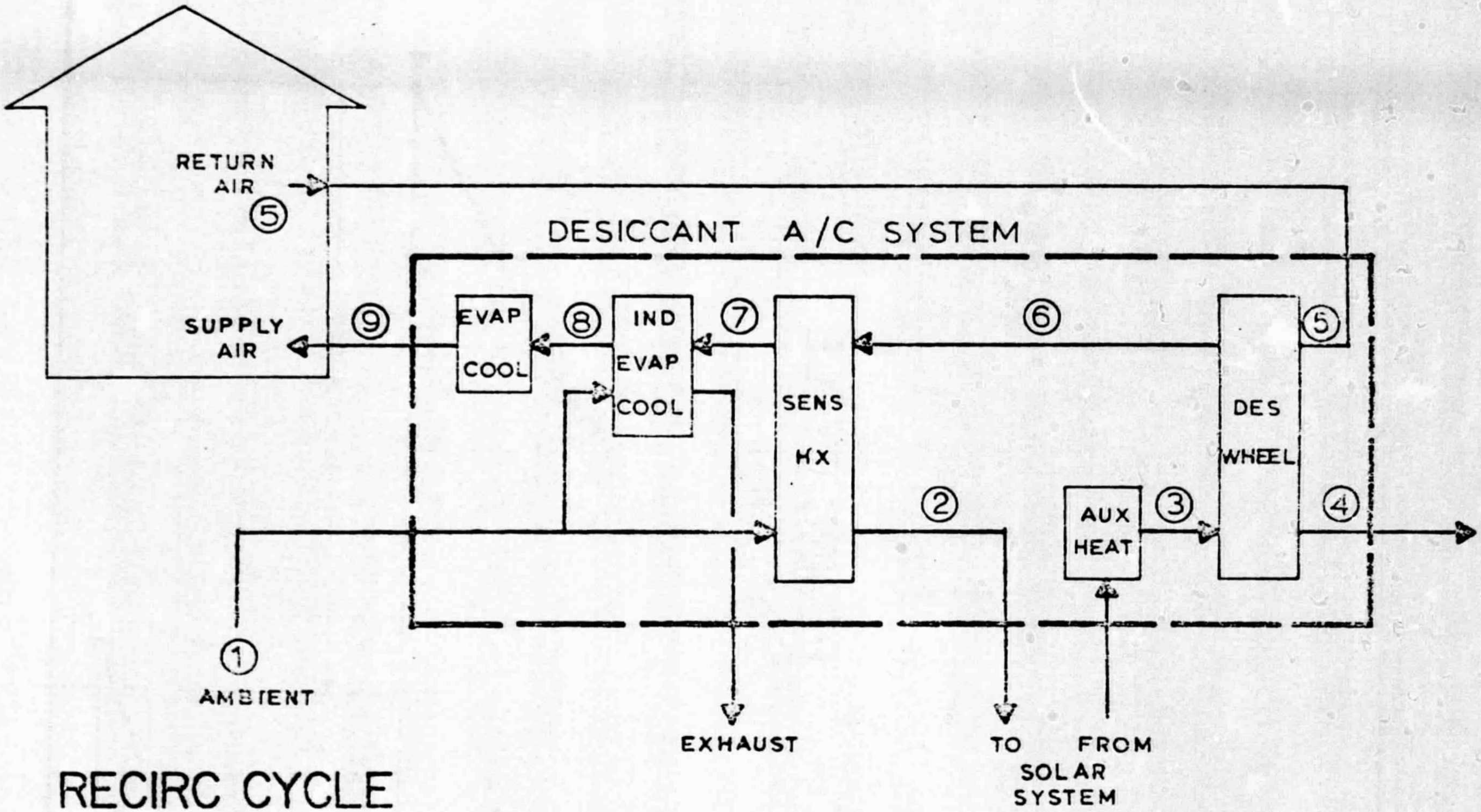


FIG. 5
RECIRC CYCLE
OPTION-1
W
$T_{WB}$
5
50%
2
3
4
AMBIENT
1
HEAT ADDED
ROOM
9
6
8
7
75°
90°
115°
$T_{DB}$
195°




RECIRC CYCLE

OPTION 2

SENSIBLE HX $\varepsilon = 0.70$
$\varepsilon = 0.85$

INDIRECT EVAP COOLER $\eta = \dfrac{T_{DB7} - T_{DB8}}{T_{DB7} - T_{WB1}} = 0.63$
$= 0.86$
$= 0.95$

**FIG. 6**

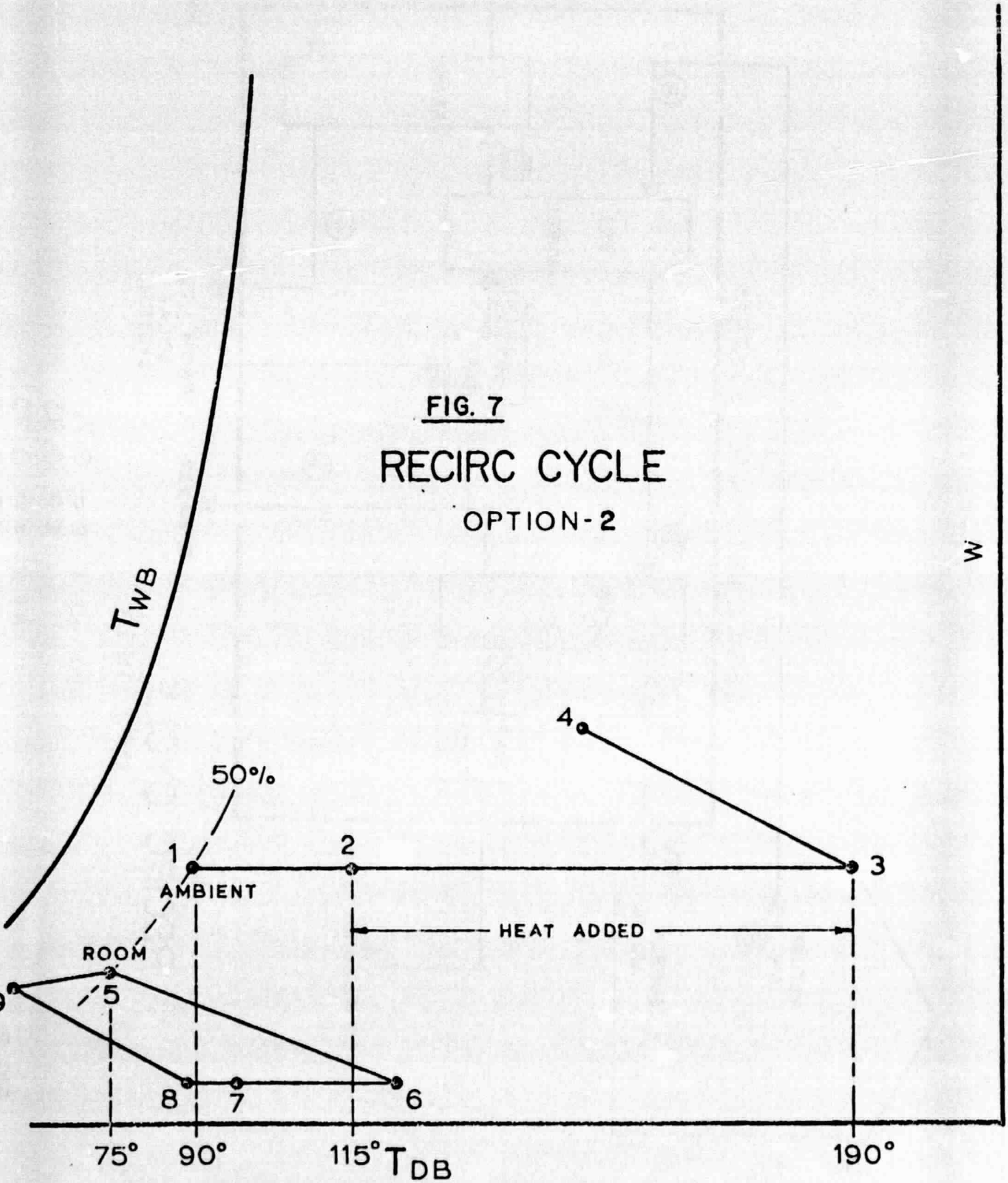
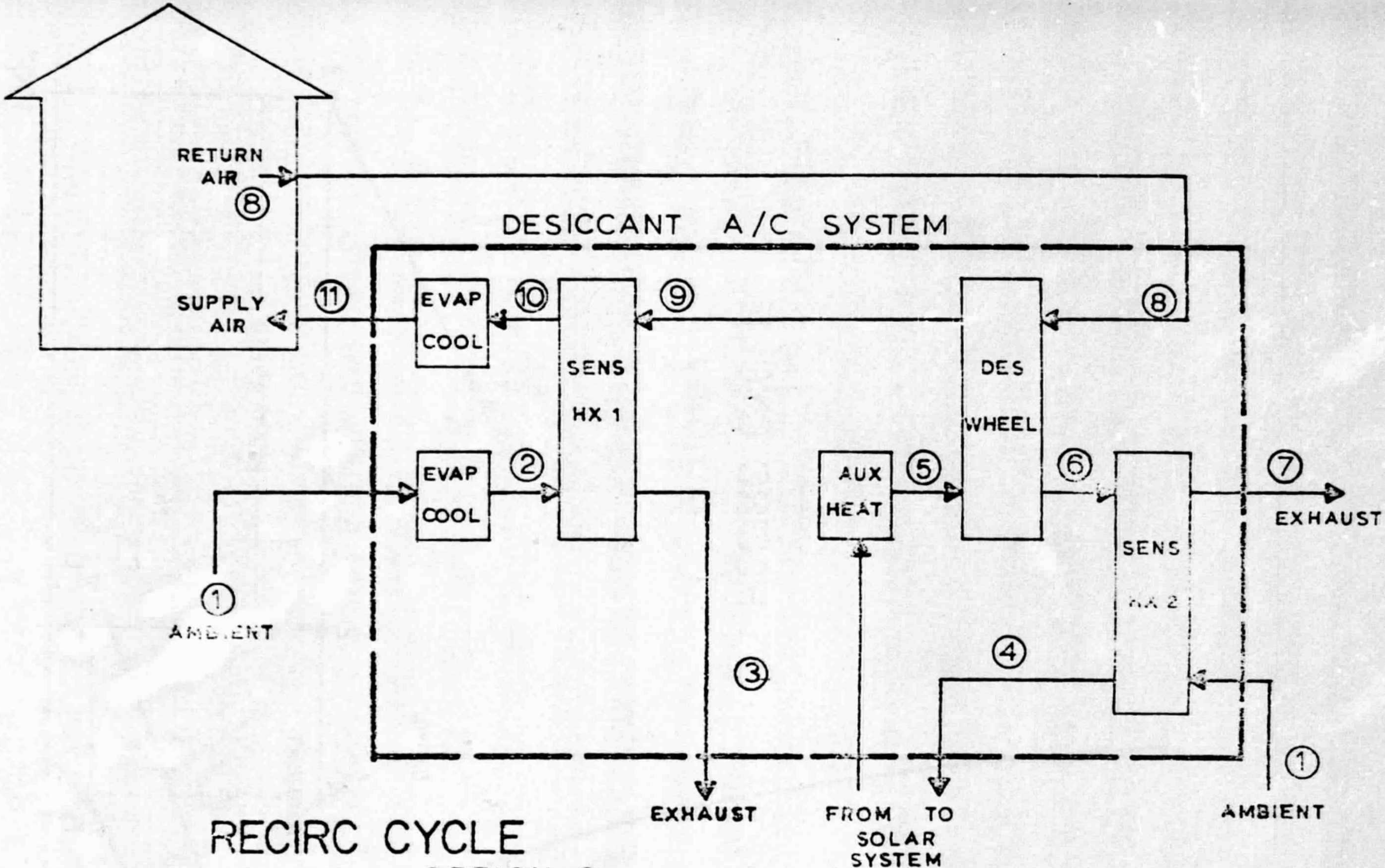



FIG. 7

RECIRC CYCLE

OPTION-2




RECIRC CYCLE

OPTION 3

SENS HX 1 $\varepsilon = 0.70$
$\varepsilon = 0.85$

HX 2 $\varepsilon = 0.70$
$\varepsilon = 0.85$

FIG. 8

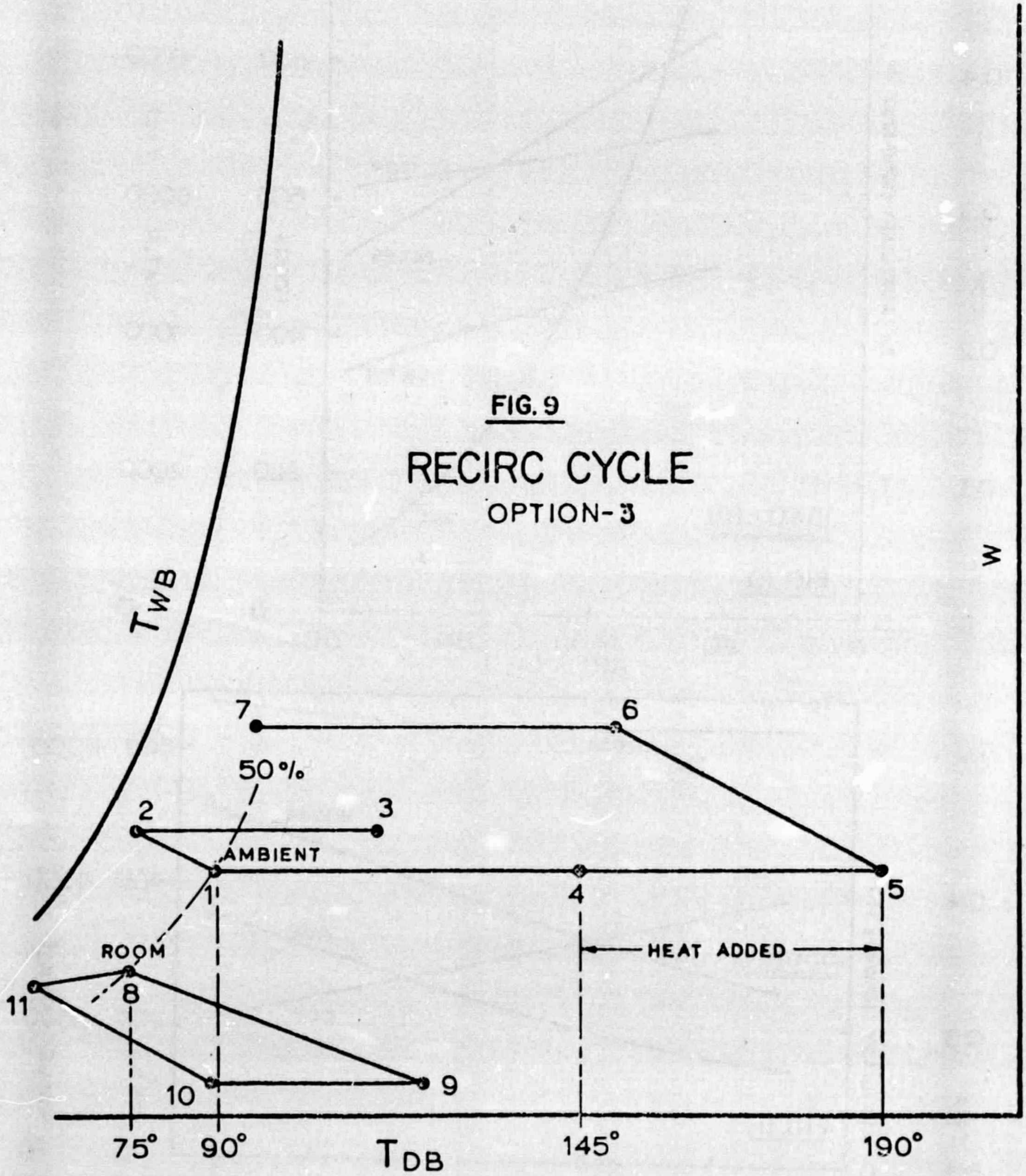



**FIG. 9**

# RECIRC CYCLE

OPTION-3

# VENTILATION MODE

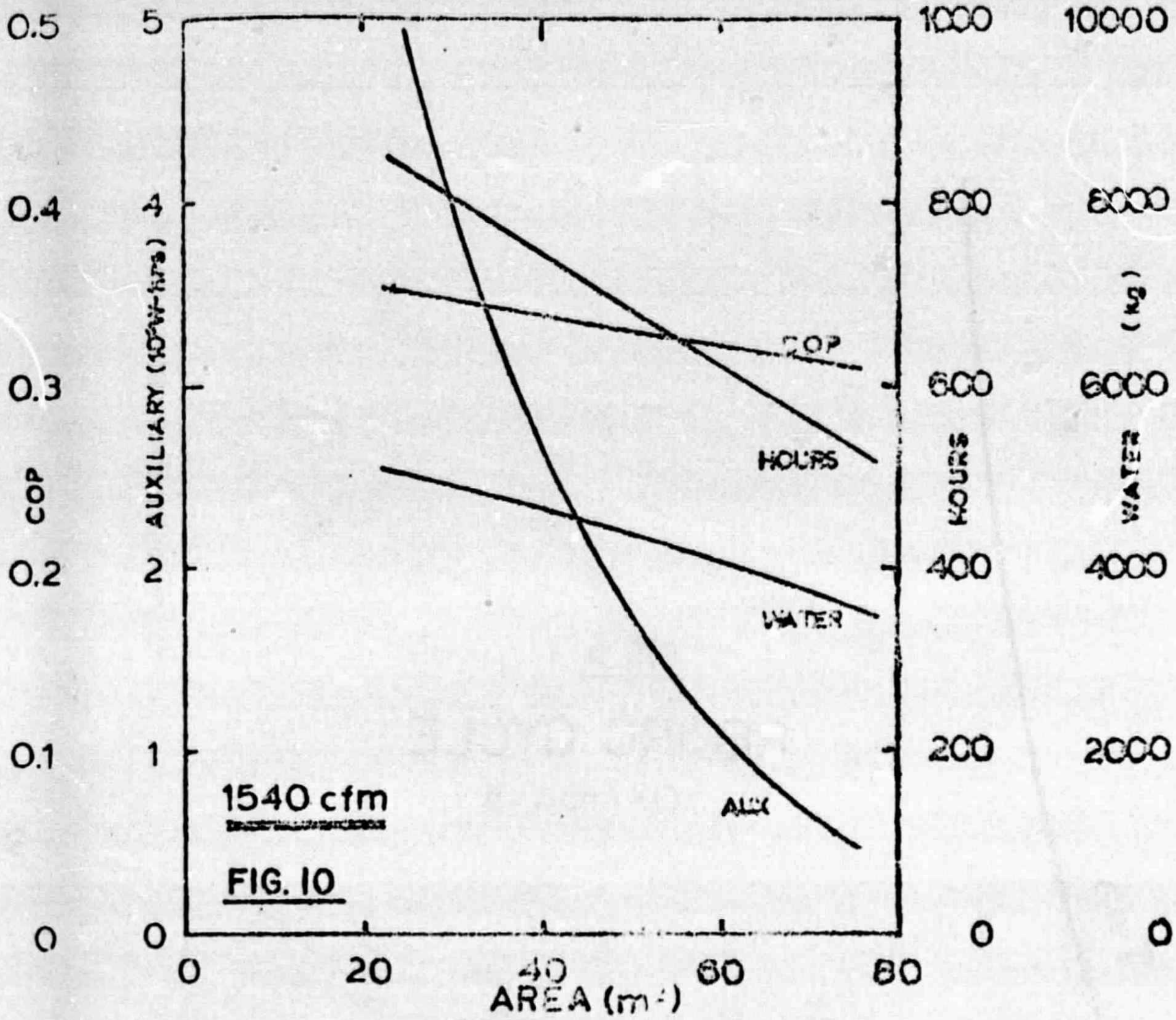



FIG. 10

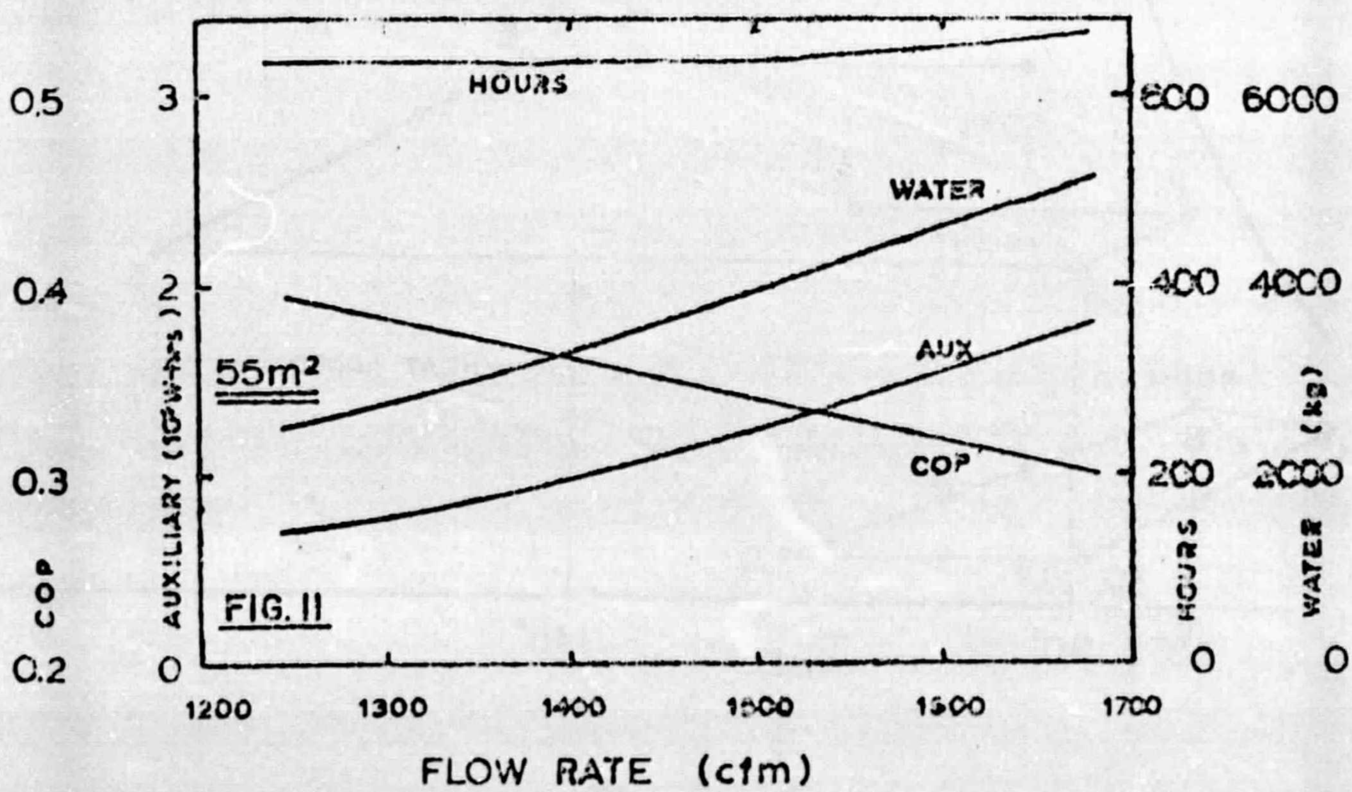



FIG. 11

# RECIRCULATION MODE OPTION 1

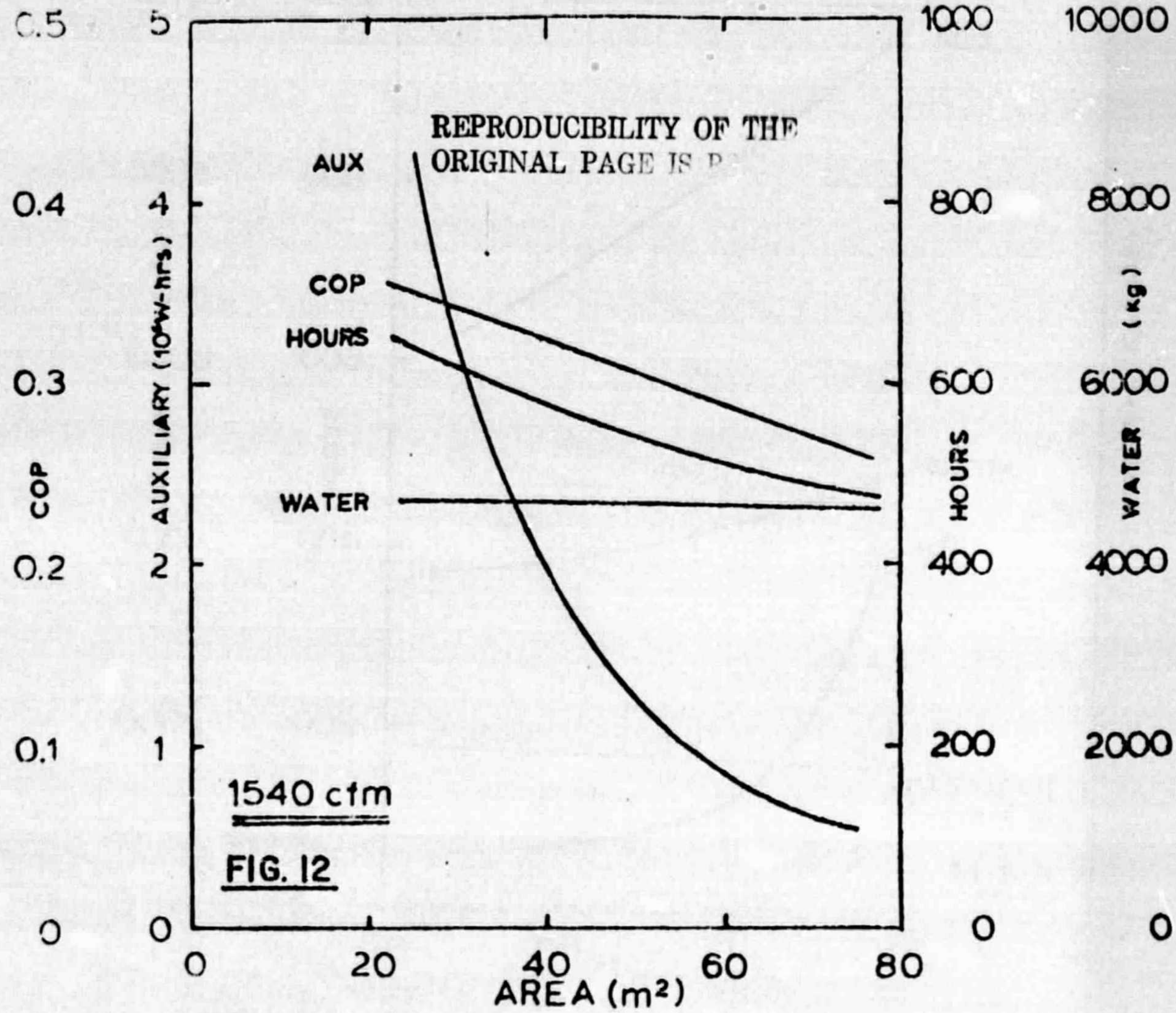




FIG. 12

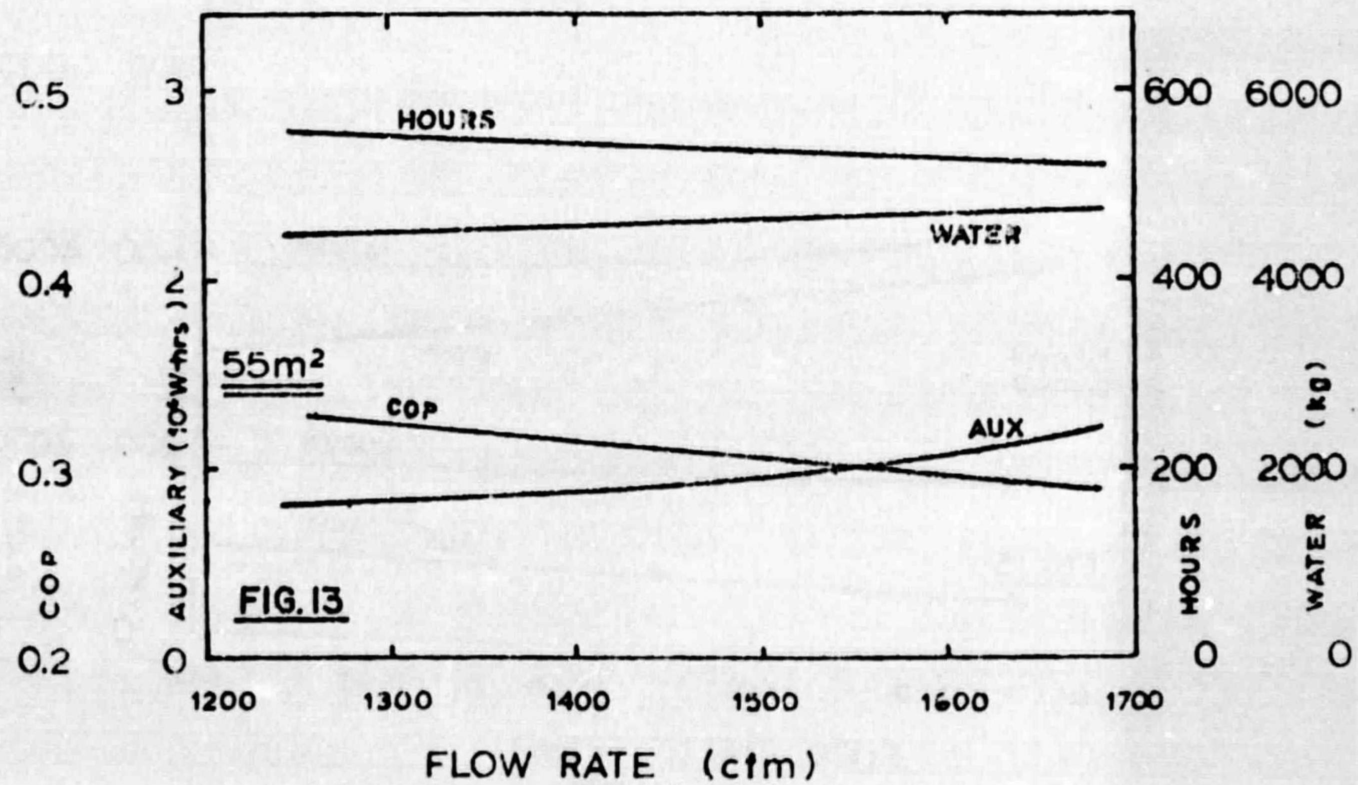



FIG. 13

# RECIRCULATION MODE OPTION 2

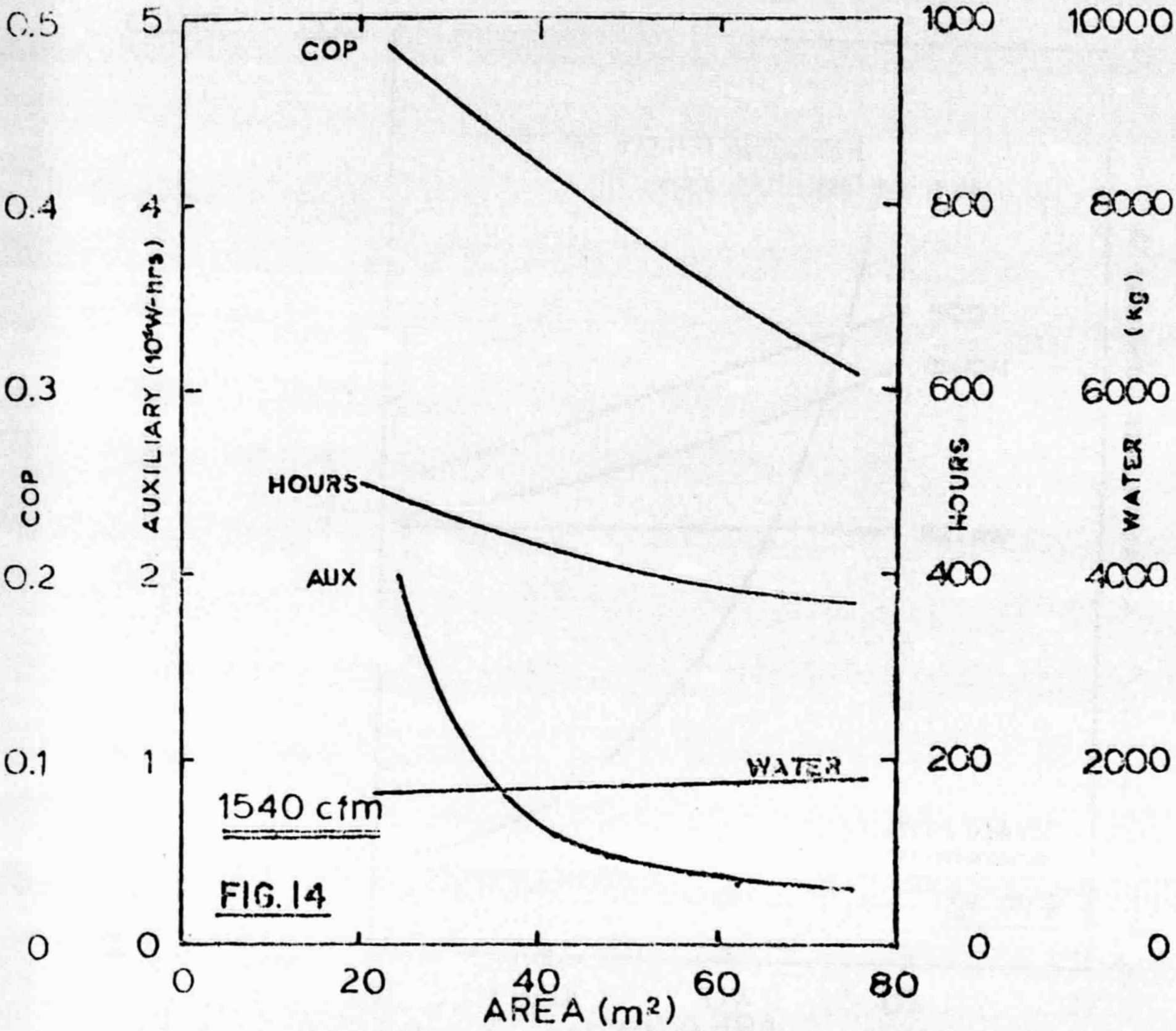



FIG. 14

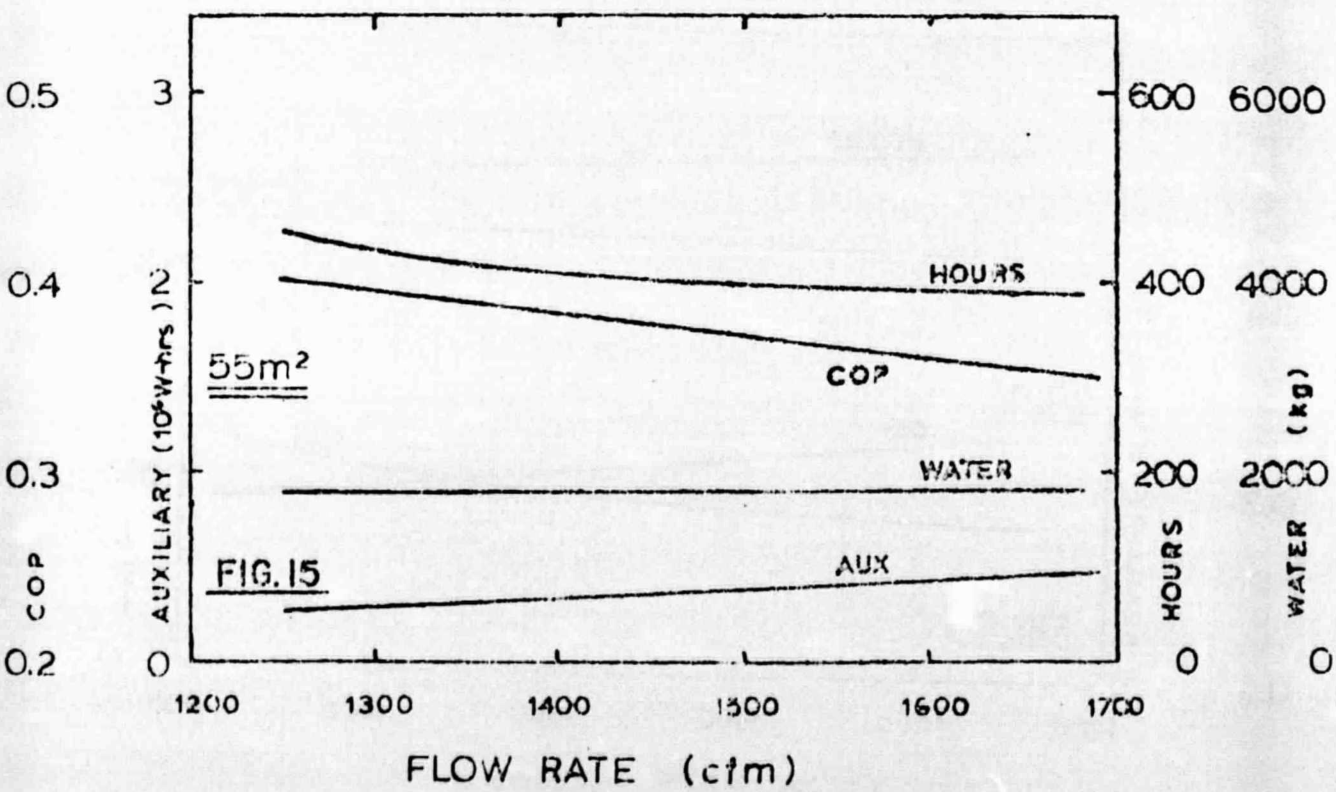
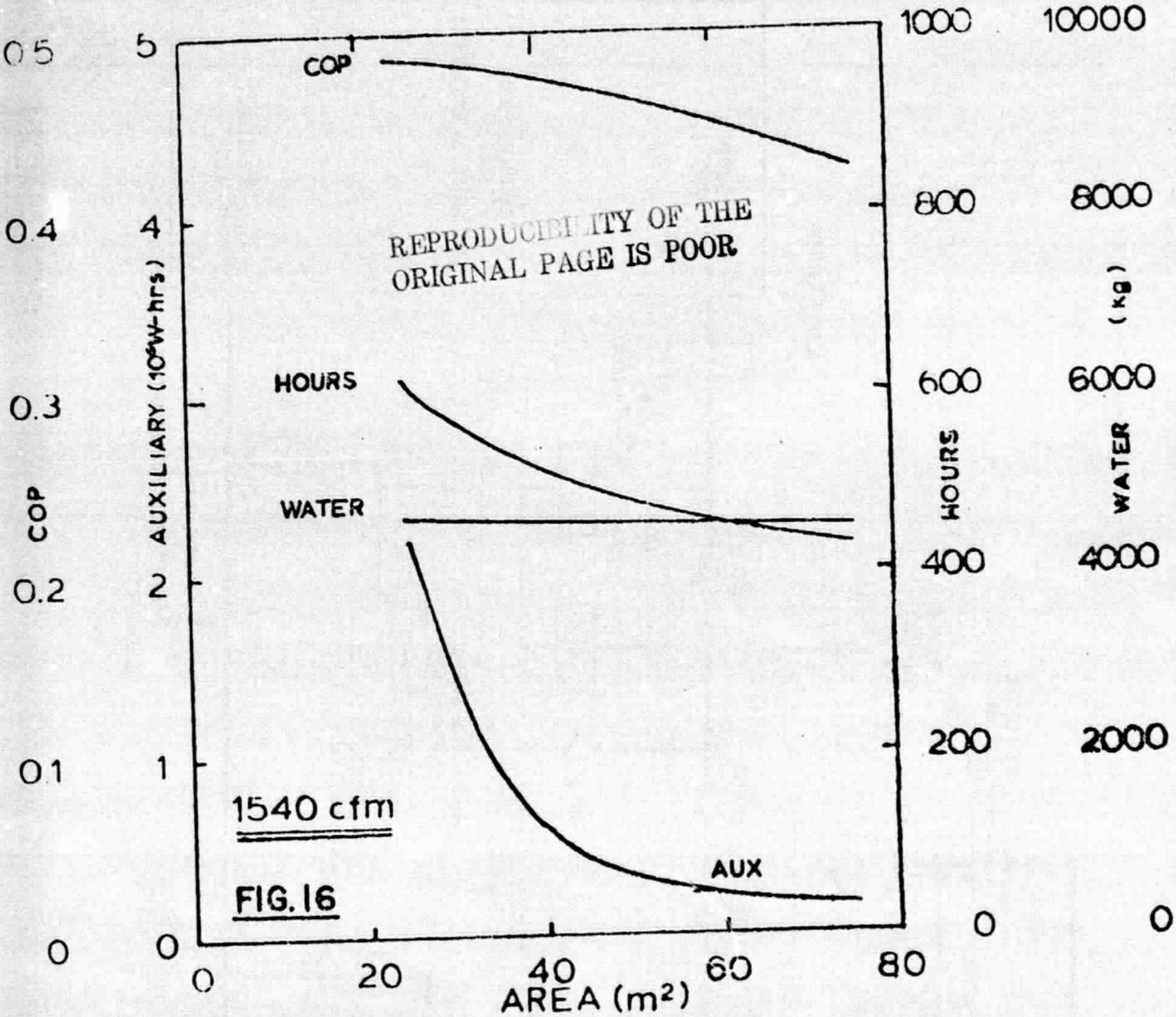



FIG. 15

# RECIRCULATION MODE OPTION 3

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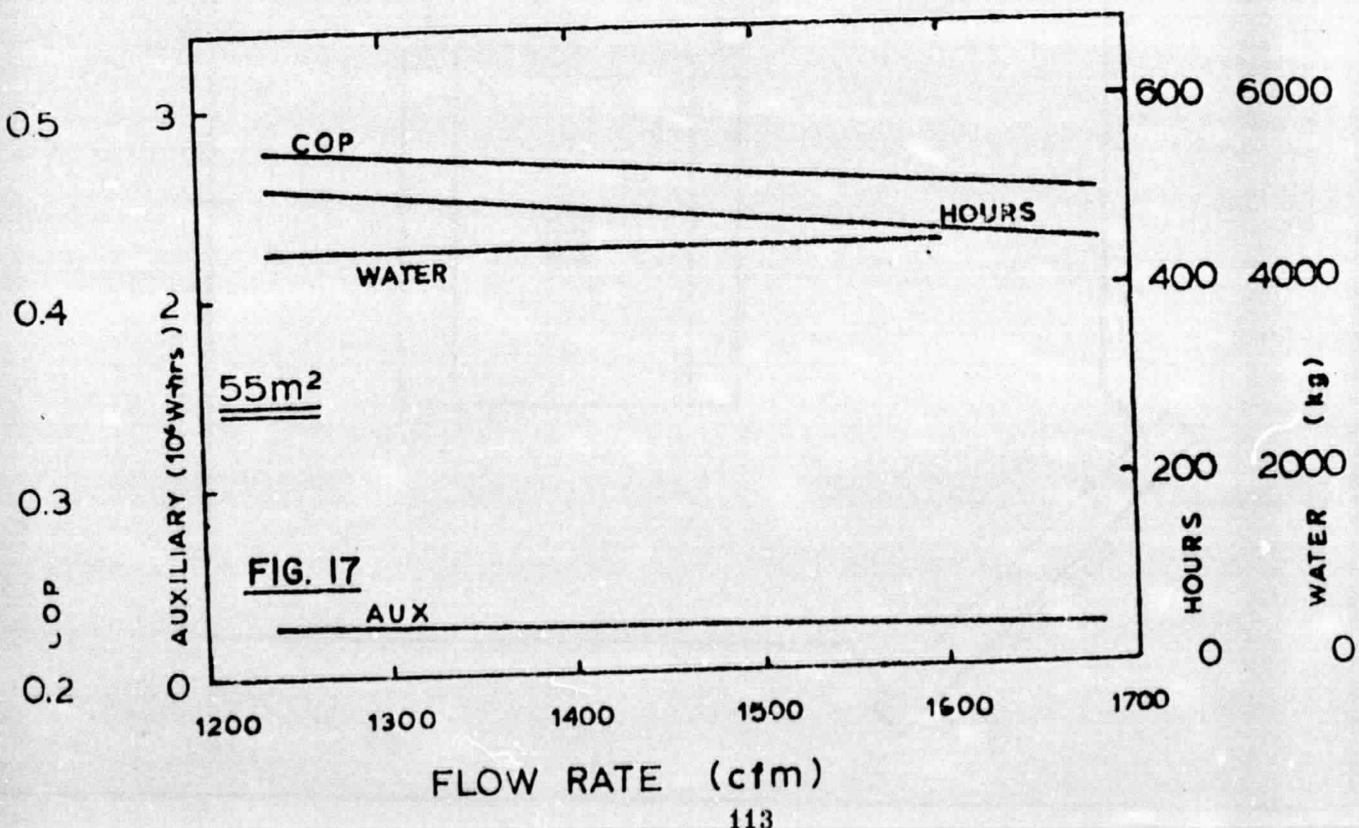


## SOLARON DESICCANT RECIRCULATION CYCLE HEATING AND COOLING SYSTEM

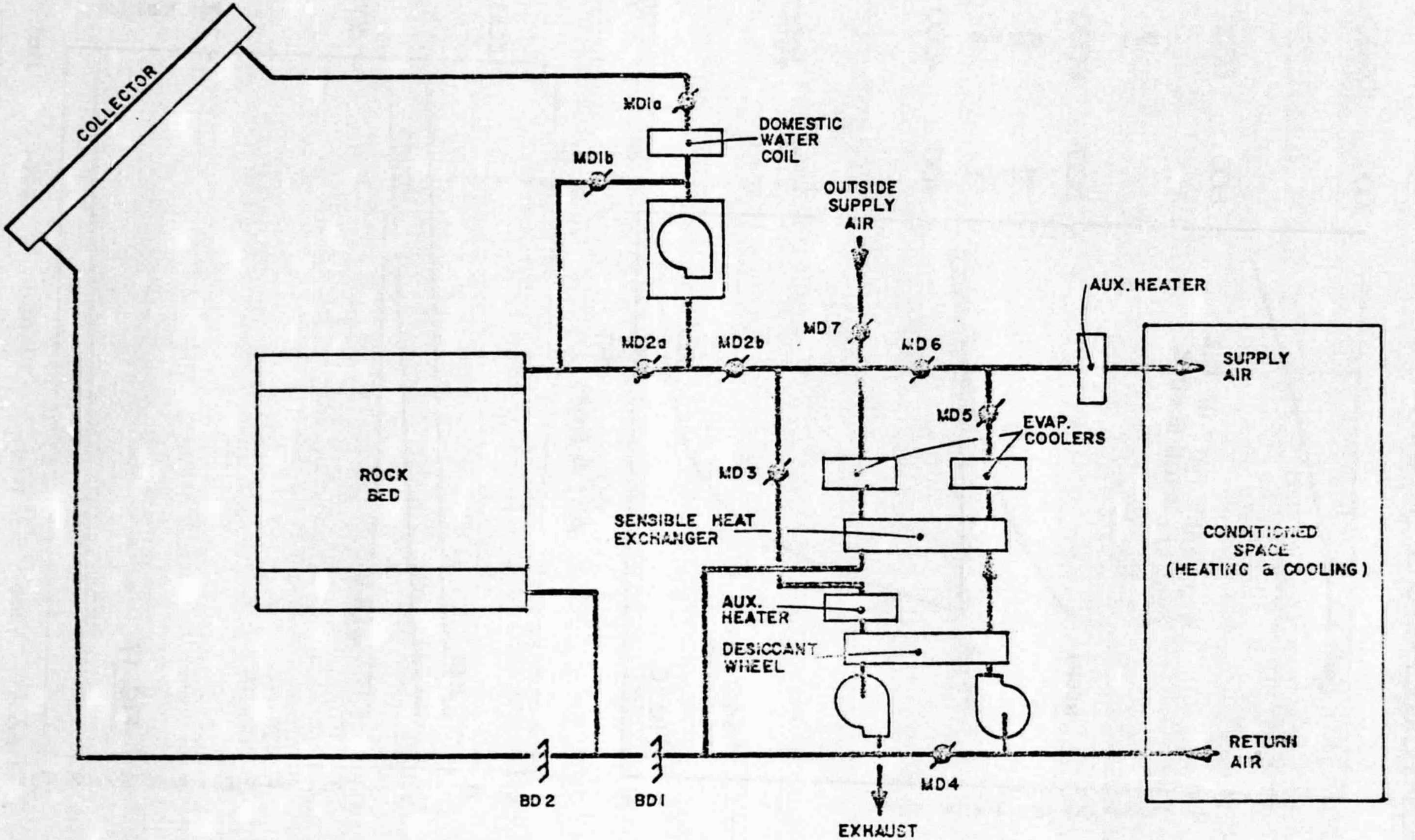



FIG. 18

SOLARON
SOLAR ENERGY SYSTEMS
Profit from the sun

# ATTACHMENT IV

(NASA CONTRACT - NAS8-32249)
AKRON HOUSE DATA SHEET

1. OWNER.......AKRON METROPOLITAN HOUSING AUTHORITY
2. LOCATION....ORCHARD PARK, NE AKRON, OHIO
3. LIVING AREA....APPR. 1200 FT.$^2$
4. ROOF ORIENTATION....FACING DUE SOUTH
   SLOPE..........45$^{\circ}$
5. CLIMATIC DESIGN CONDITIONS:

   LATITUDE.........................41$^{\circ}$ 0' N
   ELEVATION........................1210 FT. ABOVE SEA LEVEL

   WINTER DESIGN:

   MEDIAN OF ANNUAL EXTREMES.....-5$^{\circ}$F
   97½% DESIGN..............+6$^{\circ}$F

   COINCIDENT WIND VELOCITY......M (M = MODERATE, 50 TO 74$^{\circ}$ EXTREME HOURS > 7 MPH)

   DEGREE DAYS...................6037

   SUMMER DESIGN (2½% DESIGN):

   DESIGN DRY BULB..............87$^{\circ}$F
   DESIGN WET BULB..............73$^{\circ}$F
   OUTDOOR DAILY RANGE..........21$^{\circ}$F

6. HOUSE DESIGN LOADS

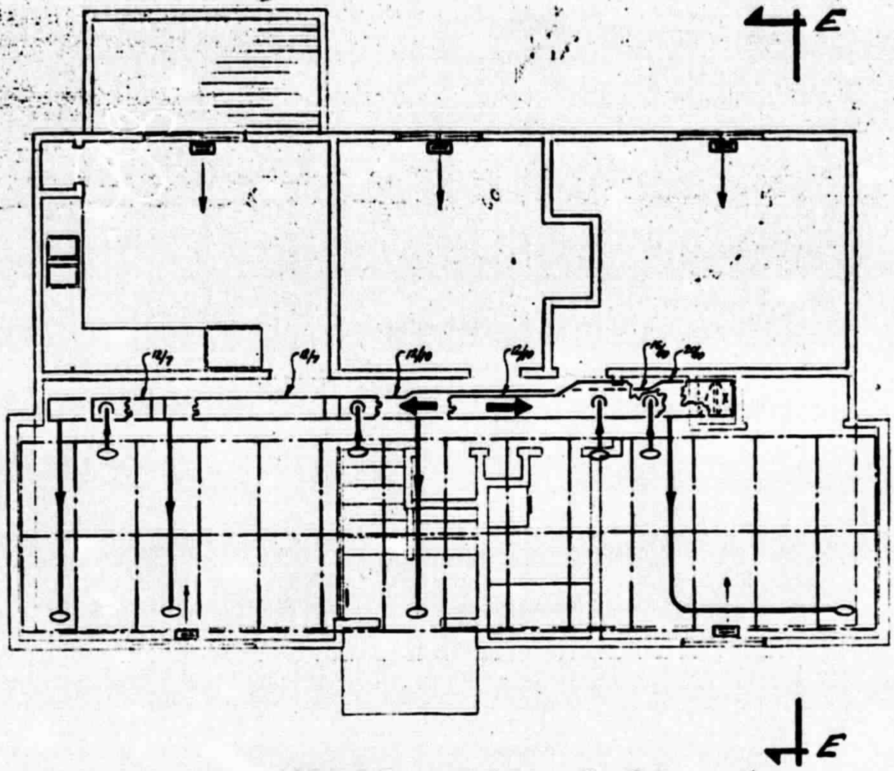
| | | |
|---|---|---|
| WINTER (HEATING) | SPACE............ | 28,759 BTU/HR. |
| | HOT WATER........ | 2,231 BTU/HR. |
| | TOTAL | 30,990 BTU/HR. |

   SUMMER (COOLING).......................21,600 BTU/HR.

7. SOLARON COLLECTORS: 28 @ 19.5 FT.$^2$ EA. = 546 FT.$^2$ TOTAL
   SOLAR ANNUAL PERCENT OF HEATING LOAD: 54.4%
   SOLAR ANNUAL PERCENT OF COOLING LOAD: APPR. 90%
   (BASED ON COLUMBUS, OHIO SOLAR DATA)

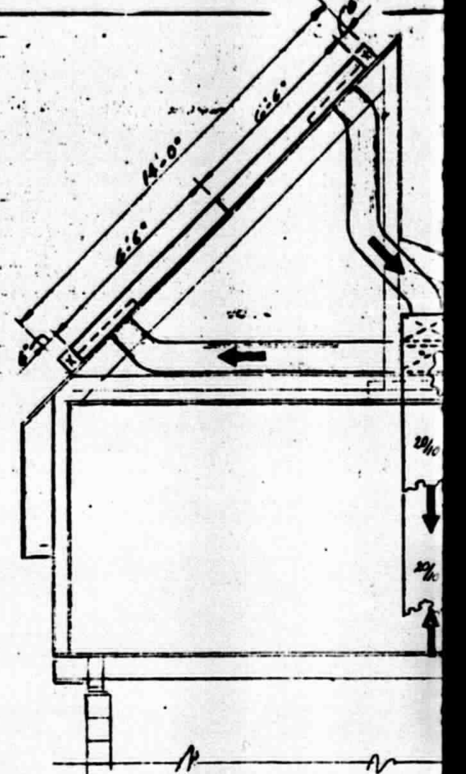
ATTACHMENT V

FOLDOUT FRAME




UPPER LEVEL PLAN

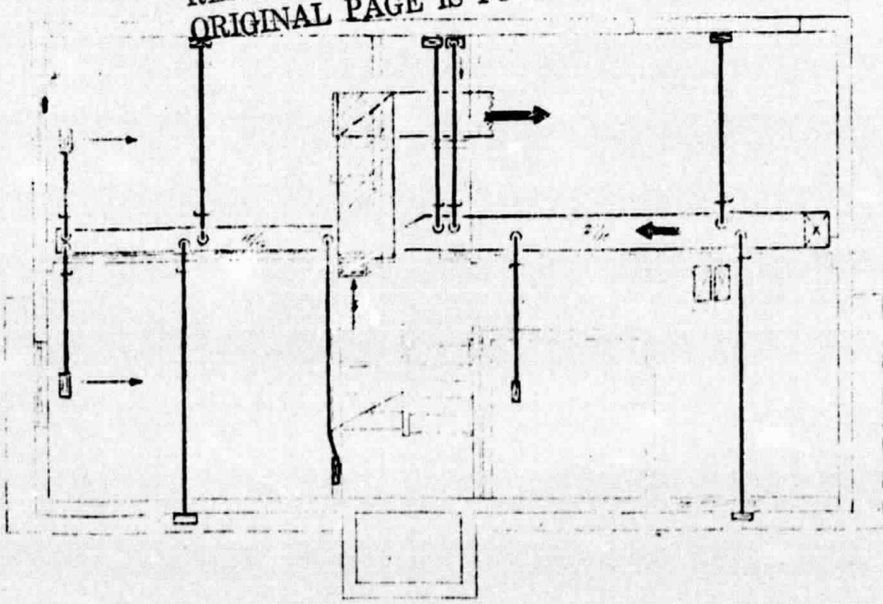
SCALE 1/4"=1'-0"




SECTION E-E

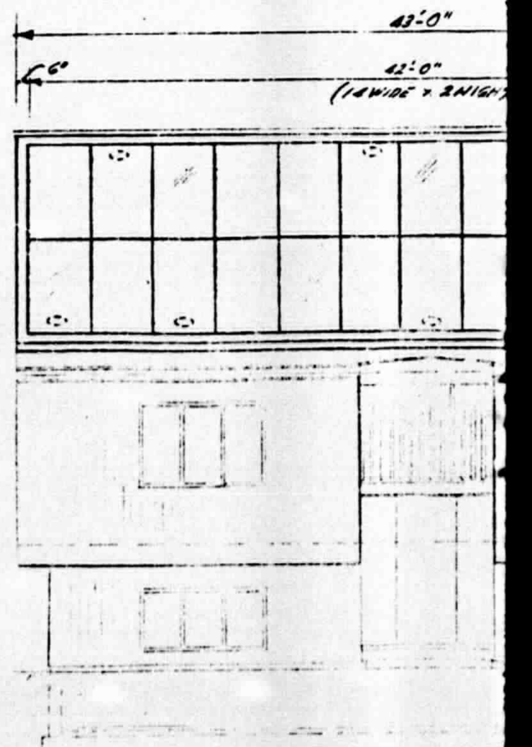
SCALE 3/8"=1'-0"





LOWER LEVEL PLAN

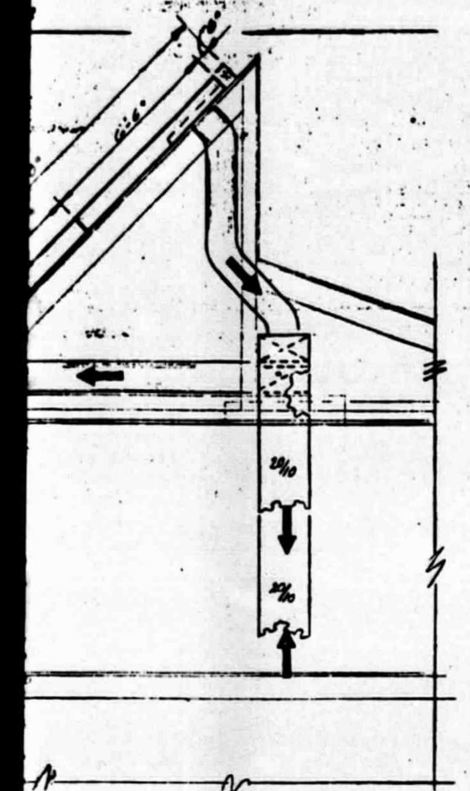
SCALE 1/4"=1'-0"




SOUTH ELEV

SCALE 1/8"=1'-0"

FOLDOUT FRAME 2



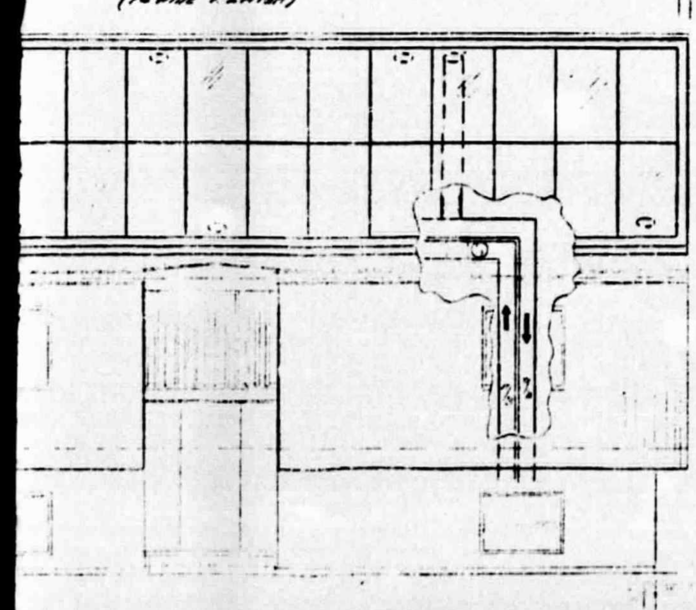

SECTION E-E

SCALE 3/8"=1'-0"

43'-0"

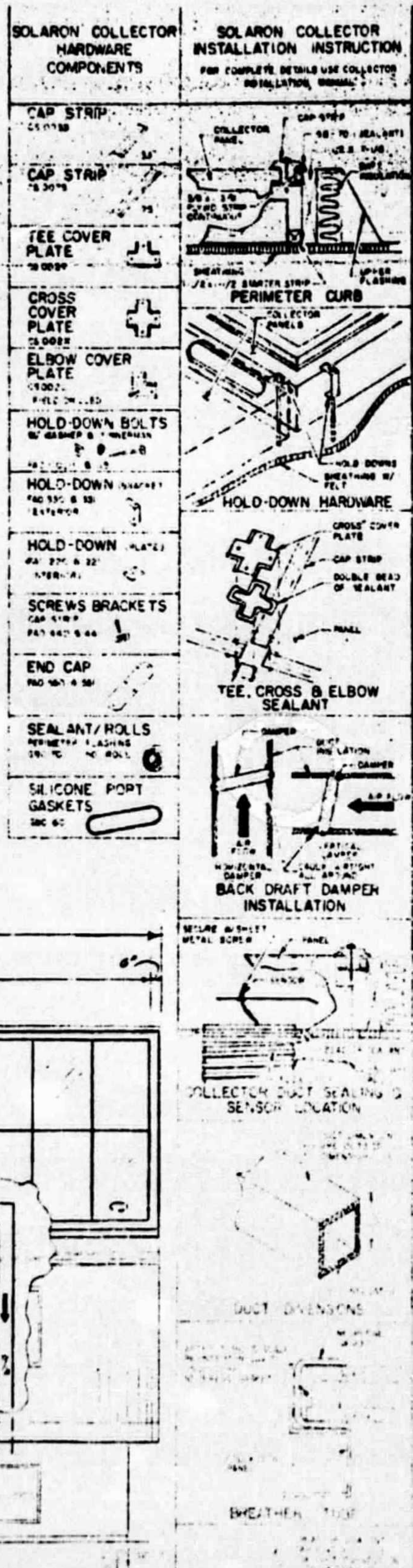
42'-0"

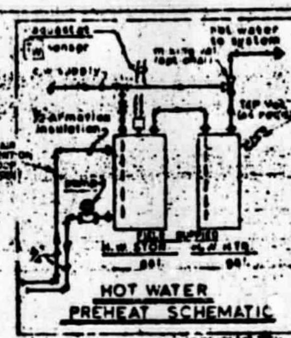
(14 WIDE × 2 HIGH)



SOUTH ELEVATION

SCALE 1/8"=1'-0"

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ATTACHMENT 5

| DESCRIPTION | DATE |
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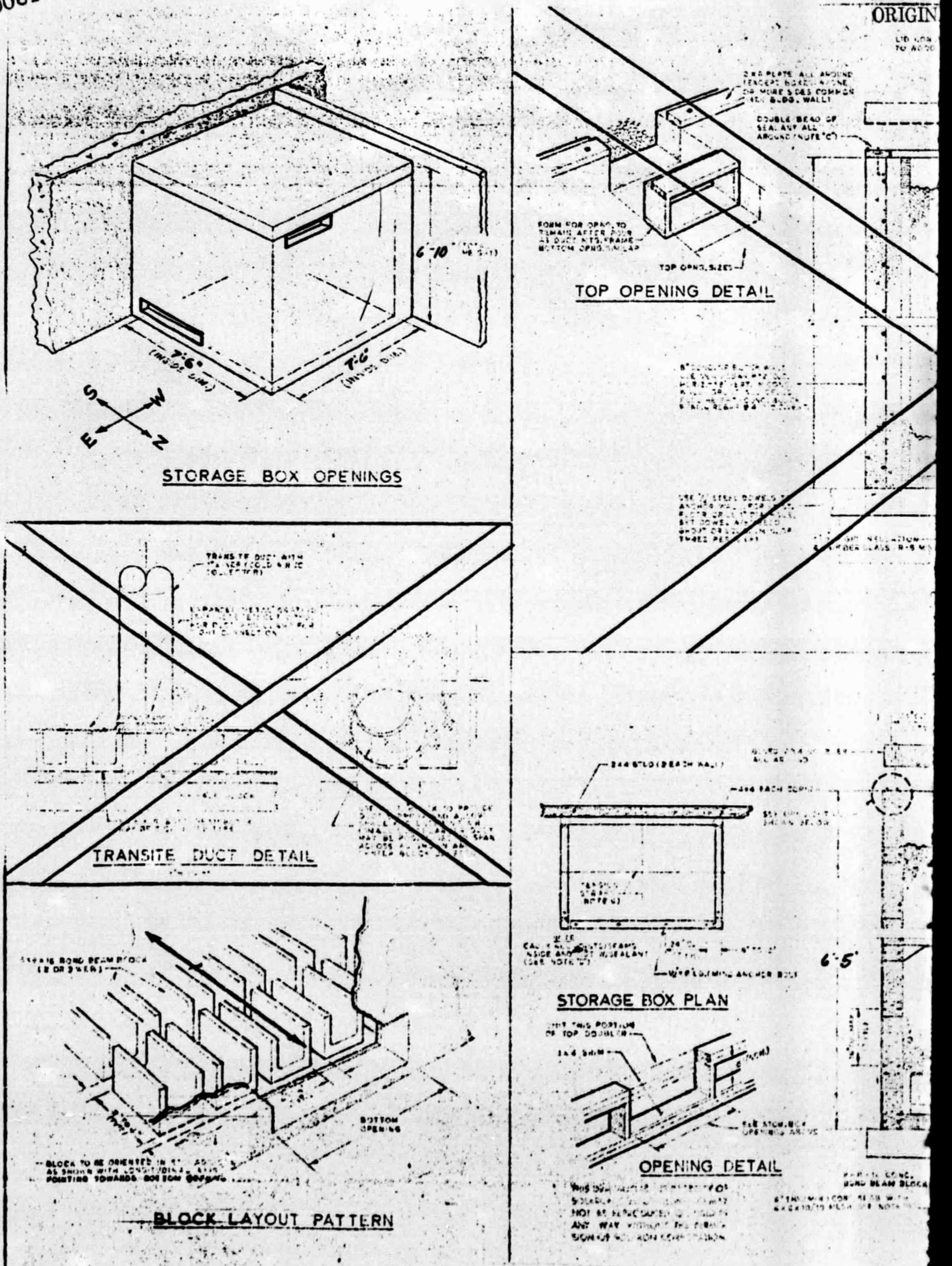
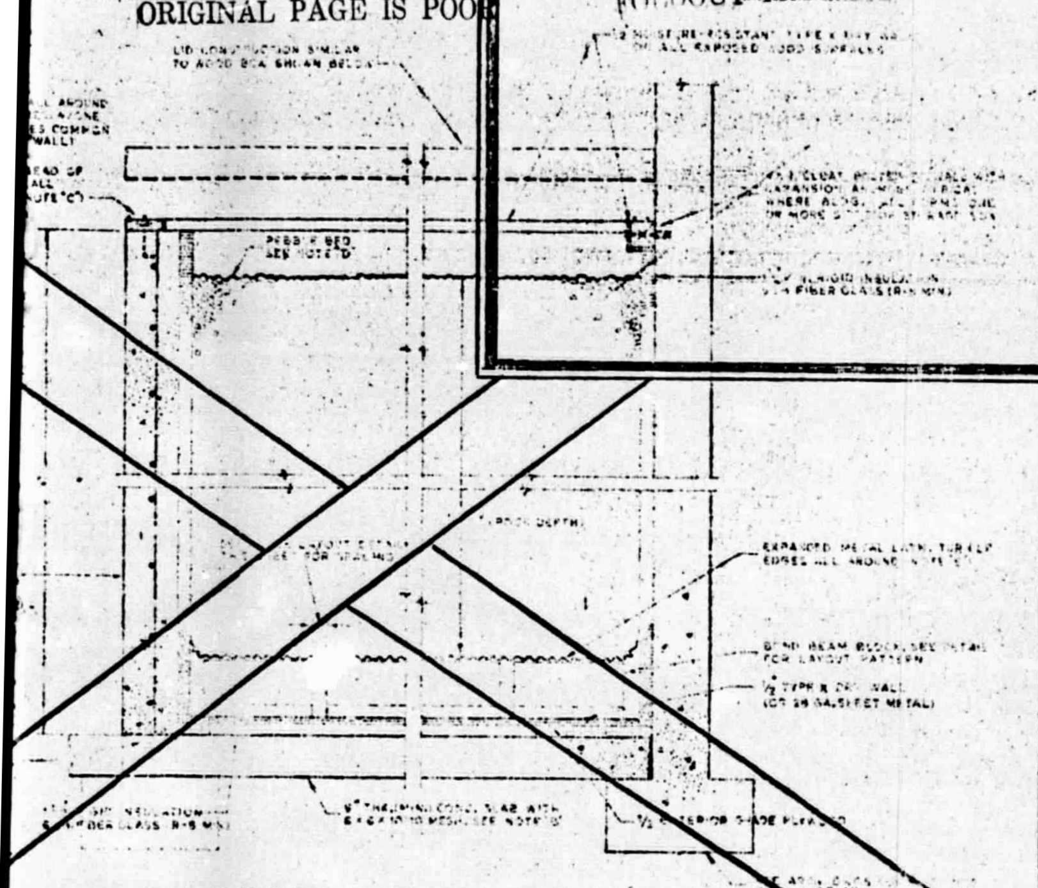
118


7'-6"
(INSIDE DIM.)
7'-6"
(INSIDE DIM.)
6'-10"
S
W
E
N
STORAGE BOX OPENINGS
TOP OPENING DETAIL
TOP OPNG. SIZES
TRANSITE DUCT DETAIL
BLOCK LAYOUT PATTERN
BOTTOM OPENING
STORAGE BOX PLAN
6'-5"
OPENING DETAIL
2x4 SHIM

FOLDOUT FRAME 2




SECTION - HEAT STORAGE BOX

CONCRETE

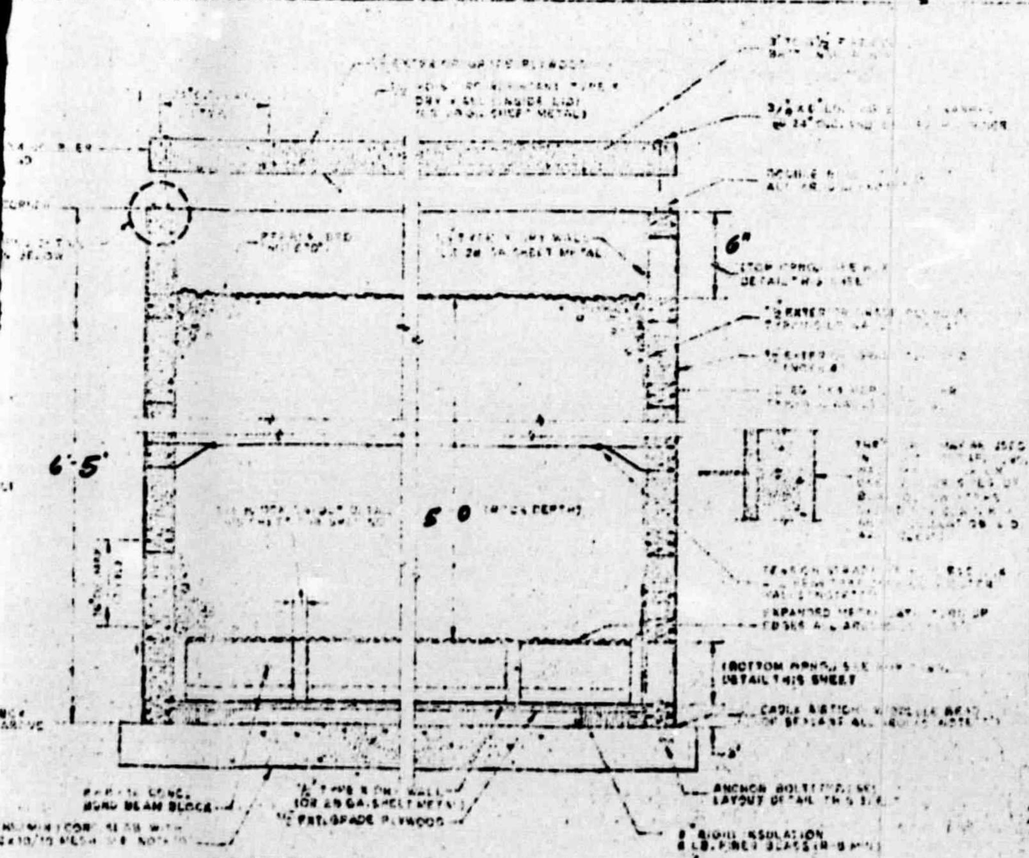
## GENERAL NOTES

## CONCRETE CONSTRUCTION

## WOOD CONSTRUCTION

281 ... 3/4 TO 1 1/2 IN. DIAMETER

14 ... /CUBIC FOOT




SECTION - HEAT STORAGE BOX

WOOD

AKRON HOUSE
ATTACHMENT 5

SOLAR ENERGY SYSTEMS
DENVER, COLO. 80222

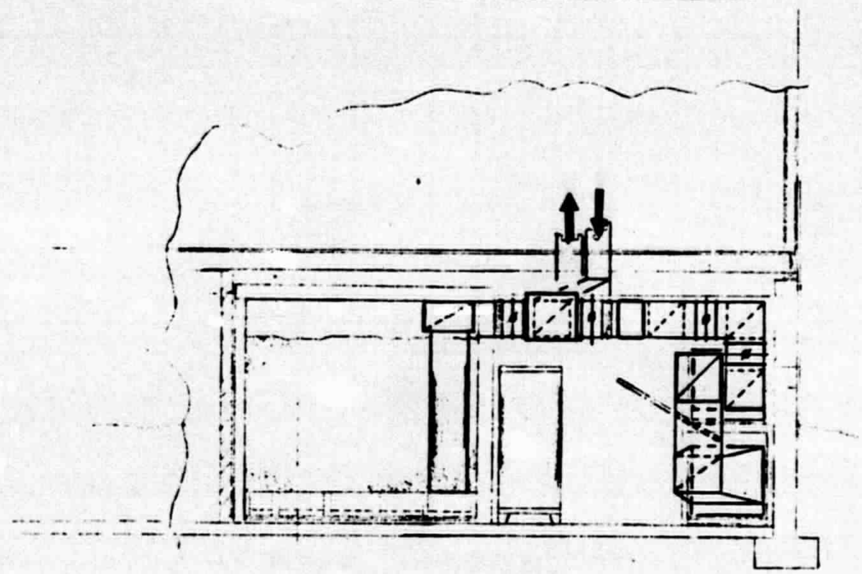
HEAT STORAGE UNIT

119

D-1002 3

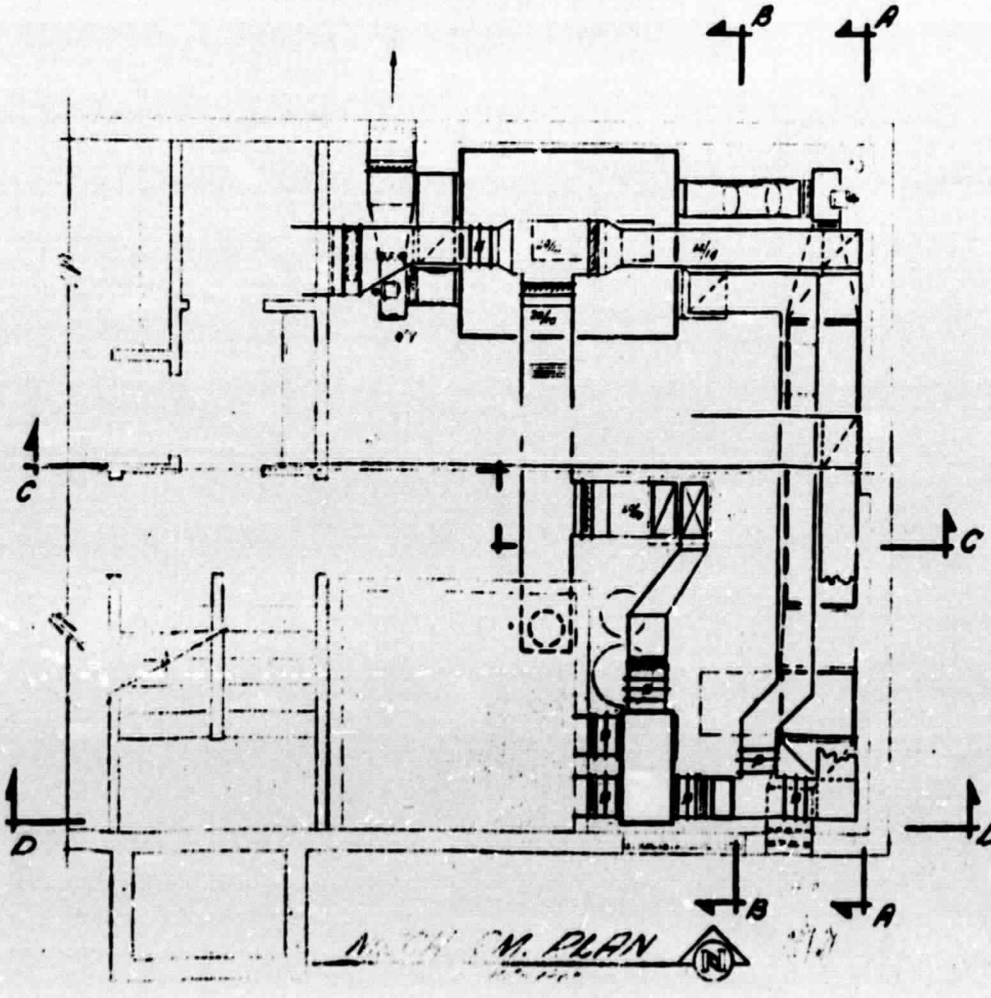
FOLDOUT FRAME

1

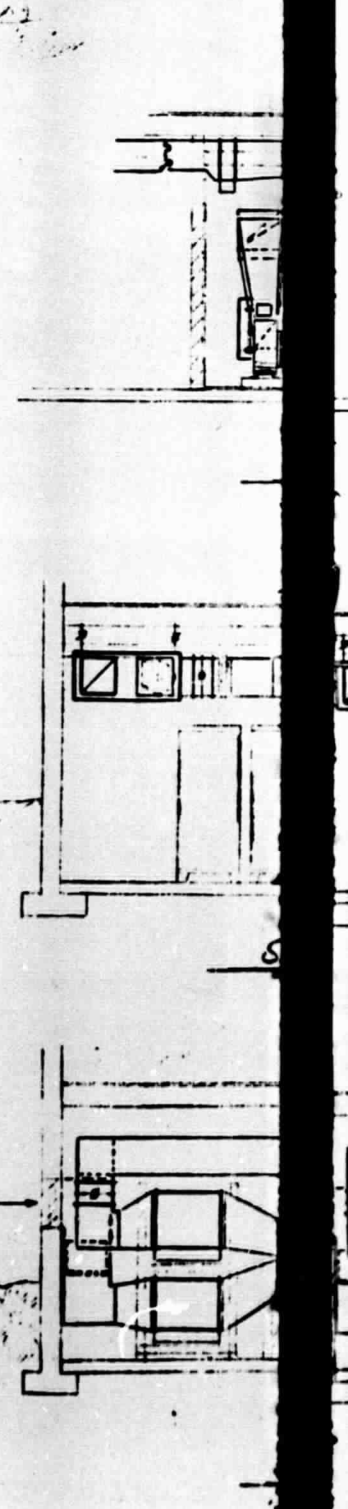


SECTION D-D

SCALE 1/2" = 1'-0"

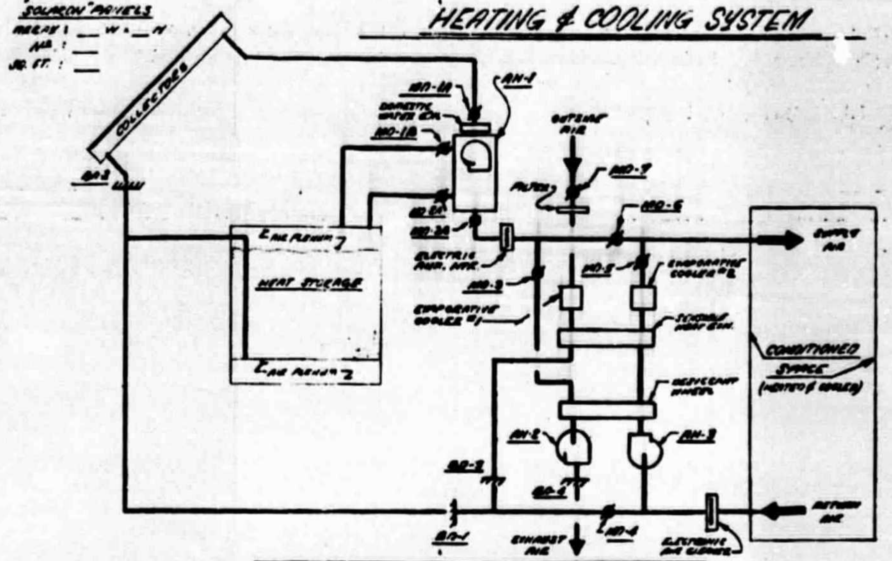



MECH. RM. PLAN



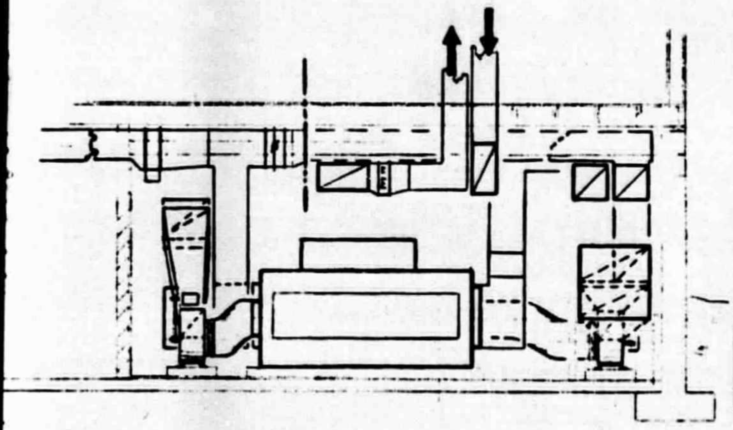
FOLDOUT FRAME 2

# SOLARON DESICCANT RECIRCULATION CYCLE HEATING & COOLING SYSTEM

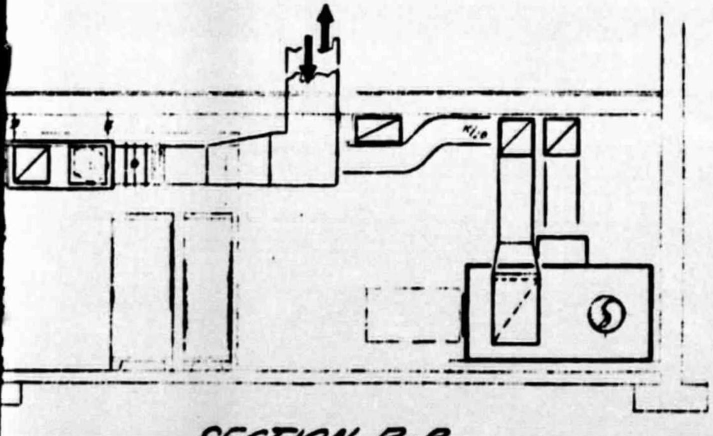



SOLAR FLOW SCHEMATIC

| | SEQUENCE OF OPERATIONS | | | | | | | | | | | | | | O = OPEN C = CLOSED | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| MODES | MD-1A | 1B | MD-2A | 2B | MD-3 | MD-4 | MD-5 | MD-6 | MD-7 | BD-1 | BD-2 | BD-3 | BD-4 | ΔP | AUX HTR. | AH-1 | AH-2 | AH-3 |
| WINTER: HEAT FROM COLLECTOR | O | C | C | O | C | O | C | O | C | O | O | C | C | | OFF | ON | OFF | OFF |
| WINTER: HEAT FROM STORAGE | C | O | C | O | C | O | C | O | C | O | C | C | C | | OFF | ON | ON | OFF |
| WINTER: HEAT FROM STORAGE & AUX. | C | O | C | O | C | C | C | O | C | O | C | C | C | | ON | ON | ON | OFF |
| WINTER: STORE HEAT | O | C | O | C | C | O | C | O | C | C | O | C | C | | OFF | ON | OFF | OFF |
| DEAD STATE | C | [illegible] | C | O | C | O | C | O | C | C | C | C | C | | OFF | OFF | OFF | OFF |
| SUMMER: REGENERATE FROM COLL. & COOL | O | C | C | O | O | C | O | C | O | O | O | O | O | | OFF | ON | ON | ON |
| SUMMER: REGENERATE FROM STORAGE & COOL | C | C | C | O | O | C | O | C | O | O | C | O | O | | OFF | ON | ON | ON |
| SUMMER: REGENERATE FROM STOR & HTR & COOL | C | O | C | O | O | C | O | C | O | O | C | O | O | | ON | ON | ON | ON |
| SUMMER: STORE HEAT | O | C | O | C | C | O | C | O | C | C | O | C | C | | OFF | ON | OFF | OFF |
| | | | | | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | | | | | |



SECTION C-C
SCALE 1/2" = 1'-0"



SECTION B-B
SCALE 1/2" = 1'-0"



AKRON HOUSE

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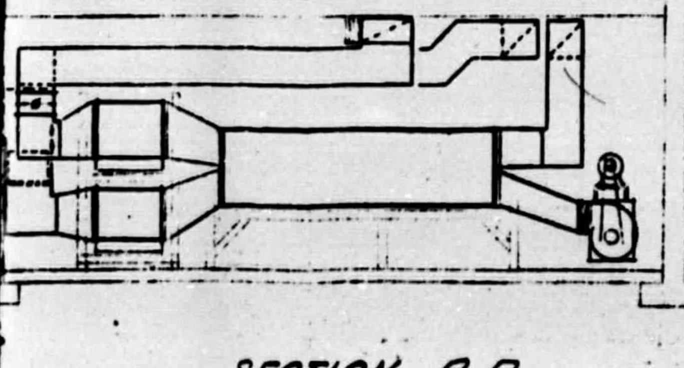

SECTION A-A
SCALE 1/2" = 1'-0"

ATTACHMENT 5

SOLARON
SOLAR ENERGY SYSTEMS
Telephone 303/759-0101
300 Galleria Tower
720 S. Colorado Blvd.
Denver Colorado 80222

150 120

Dwg. No. 13-A001

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DATE | DESCRIPTION